Heating and Ventilating

For Engineers and Contractors Controlling the Design, Specification, Installation and Maintenance of Heating, Ventilating and Air Conditioning

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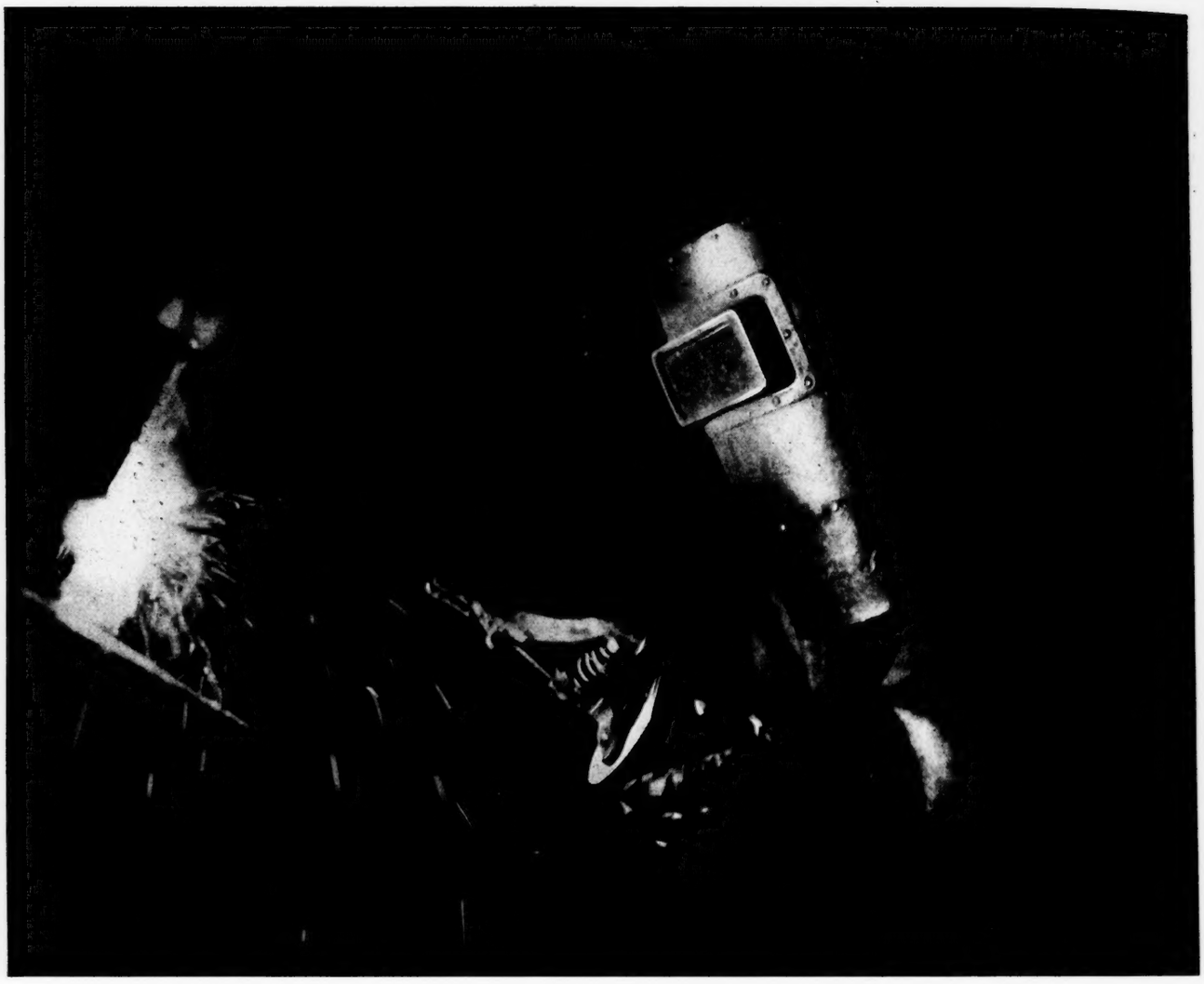
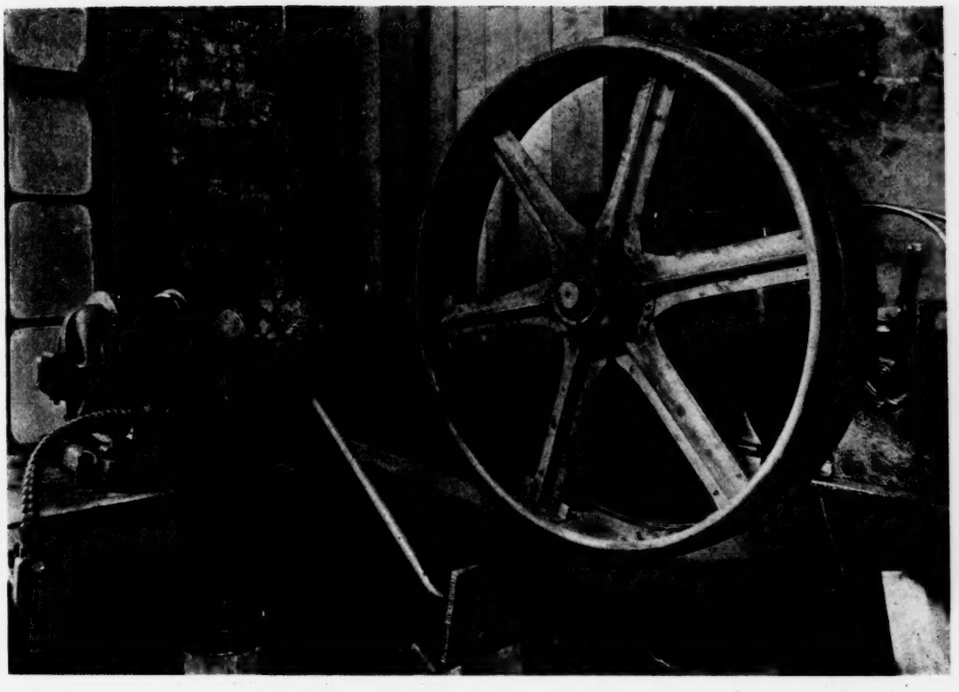


Galloway

" . . . . welded piping reflects important economic advantages to both owner and contractor . . . . the technical advantages offered by welded piping grew with the proofs that technically ideal piping can be approached easier by means of welding than by other methods."

Power Drive for Industrial Ventilating Systems

By ROBERT W. DRAKE†



Belt-driven blower for factory ventilating system

VENTILATING engineers whose experience with factory ventilation has extended over a period of years do not need to be reminded of the fact that the performance of industrial ventilating systems has a characteristic tendency to fall off with the passage of time. There is a characteristic decrease in the volume of air moved, with no change whatever in the manufacturing conditions, the duct system, or the fan equipment.

When a client states that his system was very satisfactory for a while after installation, but that conditions are becoming steadily worse these last few years, we tell him that he appreciated the great improvement at first but has become unduly critical as the memory of the original conditions became less distinct. In part we are right. In most cases, however, there is a decided falling off in performance as well.

My first experience along this line had to do with a system installed for the ventilation of a number of rooms in which painted wood and metal parts hanging on trolleys were air dried. This was many years before the introduction of ovens with continuous moving conveyors. The system involved three fan motors, a 25 hp. driving a double fan, and a 15 hp. and 10 hp. each driving single fans. After installation, simple tests were run to determine the loads on the motors. As time went on there were occasional complaints that conditions were "becoming no better fast."

They were attributed to the psychological factor mentioned above. No real attention was given to the matter until three years had elapsed. Then when an occasion arose to install some 50 new motors, instructions were issued to all motor inspectors to turn in lists of all motors under their charge which had never operated warm.

The intention was to replace these motors with smaller sizes, use the motors thus displaced in the new installation and thus improve the plant power factor. The three motors driving the fans were in these lists. They were tested for horsepower; comparison with the original tests made after installation showed a decrease in power consumption of over 30%. Tests of fan suction showed no change. The fans, which were dirty, and runners were cleaned. The ducts contained no considerable deposits or "snow drifts" of foreign matter. The matter was more or less of a mystery. Finally the 10 hp. motor was installed in place of the 15 hp., the 15 hp. in place of the 25 hp. and a new 7½ hp. was substituted for the 10 hp.

About three years later another new installation involving 50 motors offered an opportunity to shift underloaded motors and lists of cold motors were once more collected. The same three fans were once more

†Mechanical and electrical engineer, Chicago.

reported. Tests showed the same suction as read at the time of installation, but a further decrease in horsepower of decidedly more than 30%. In other words, the power consumption of these fans had decreased more than half in six years.

Once more a little attention was paid to the matter; the fans were cleaned, the screens installed over the suction inlets to prevent the entrance of lunch papers, banana peelings and dirty waste were brushed thoroughly, and such points in the ducts as could be inspected were examined for "snow banks" of dirt. Nothing serious was reported so once more the two smaller motors were moved up and a new 5 hp. unit replaced the 7½ hp. motor on the smallest fan.

Again, after a period of years a list of "cold" motors was compiled. Again all three of these fans were on the list. Similarly, a number of other ventilating fans in other locations. This time more thought was given to the matter. It was recognized that since the present speeds were identical with those recorded at previous tests, and since there had been no decrease in the suction at the fan, but rather a slight increase in some cases, the only possible explanation was decrease in volume through increase in duct resistance. The ducts were carefully examined. There was not enough reduction in cross section to account for any considerable volume decrease. However, all the trunks and branch ducts had developed a flannel or plush-like surface of lint fibres and dust, adhering perhaps through some property of the paint vapor passing through them. This pipe surface presented a far greater frictional resistance to the air than did the smooth galvanized metal surface of the ducts when new.

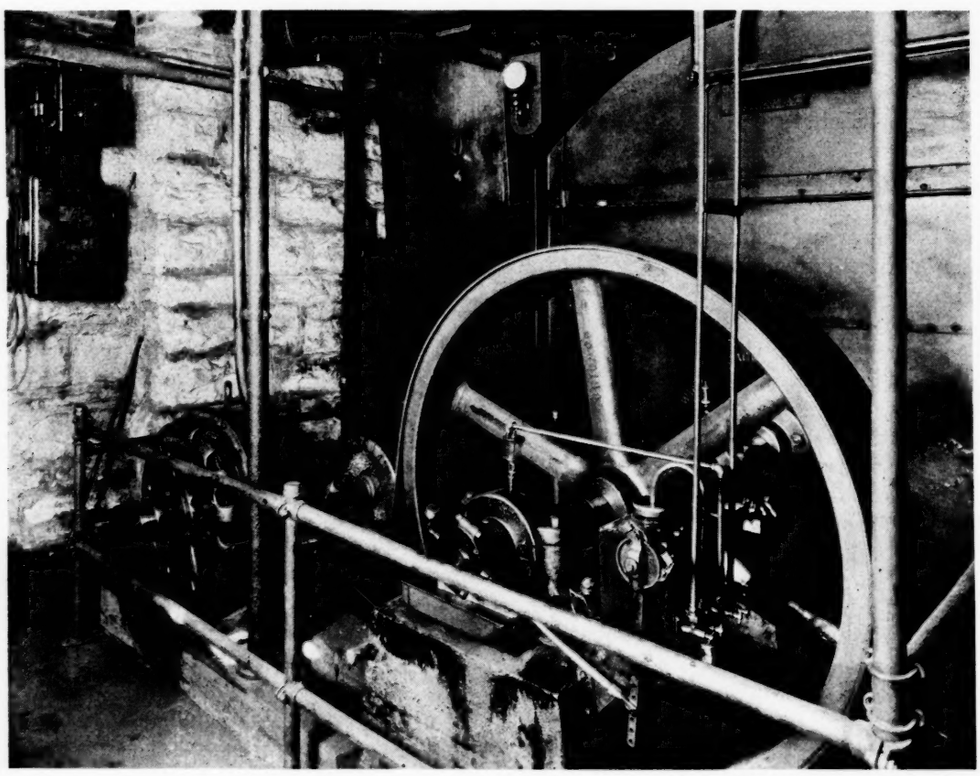
Such trunk ducts as would pass a man were brushed out. An attempt—largely unsuccessful—was made to clean the smaller branches by dragging burlap sacks filled with excelsior through some of them. Cleaning with an air hose and by rapping the pipes while the fan was in operation was also tried. The coarse (½ in. or ¾ in.) wire mesh over the suction inlets was carefully brushed and the fan vanes and housing were cleaned with a wire brush.

Another test showed an increase in horsepower, but by no means the increase expected. Evidently the cleaning of the branches, too small for a man to enter, was relatively ineffective. The greater part of the duct resistance was in the branches, due to the fact that the velocity in the branches is comparable to that in the mains, while their area was far smaller. Consequently, the cleaning of the mains accomplished comparatively little. This time motor pulleys were changed and the fan speed increased to the limit allowed by motor heating, about 25% above the rated capacity of these particular motors.

In later years there have come to the writer's attention many other cases of ventilating and exhaust systems which failed to give adequate performance after the lapse of a few years. The cause is not always the same. Sometimes the causes are ludicrous when we find them, and in no way the result of increased duct resistance or decreased fan performance, as, for example, a runner with a centered hub reversed after removal for a change in motor bearings or for fan cleaning, or the interchange of the right- and left-hand runner of a double fan when both have offset hubs and have been removed for the same purpose.

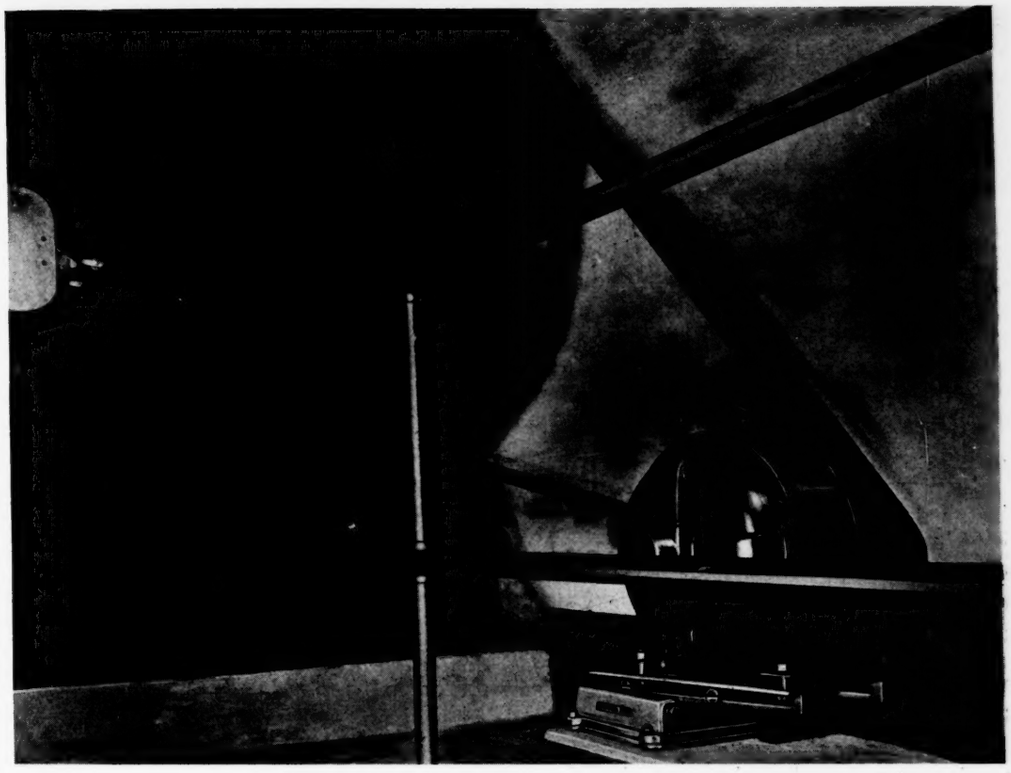
Nevertheless, in a considerable number of industrial exhaust and ventilating installations a moderate change of fan speed would be advantageous at one time or another during the life of the job. Fortunately not all such changes involve a speed increase.

Among the most frequent in occurrence are the in-



Fan room of a mid-western industrial plant.

Belt-driven blower for a well ventilated industrial plant.



stallations where the ductwork is full of bends, offsets and elbows to avoid obstructions already existing. There may be a number of changes from round to square pipe and back again. Estimates of resistance through such ducts are of necessity rough. The designer, anxious to assure a satisfactory job, estimates resistances liberally. When the job is in operation it is evident that decidedly more air is being moved than is needed. A 10% or 20% decrease in fan speed will give ample ventilation, save the client a third to more than half the power required to drive the fan, and substantially reduce the heating load.

Then there are the cases where changing manufacturing methods increase or decrease the number and size of suction outlets moderately, where change in process changes the amount of suction necessary, where the installation of a duct arrester or an air filter increases the suction or pressure desired. Where the changes required are not too great, a moderate variation in fan speed will ordinarily give a serviceable job. Where duct resistance increases through the years and thorough cleaning is impractical—or the client is convinced that it is impractical, which is often the same thing in effect—the system performance may be maintained satisfactorily by a somewhat perfunctory cleaning and a moderate increase in fan speed every few years, say, 5%, or at most 10%, increase in speed, corresponding to 10% and 20% increase in pressure or suction.

With belted fans such changes may be effected simply and at small expense, with direct connected fans, moderate changes in speed are impossible when the fans are driven by means of ordinary types of a.c. motors. The effect of very small speed increases or moderate decreases in speed in direct connected induction motor-driven fans may be accomplished by the installation of special runners of slightly different diameter than standard.

While offhand we might think that the substitution of motor of higher or lower speed would be a solution in many cases, the fact is that this is seldom practicable. The steps between a.c. motor speed are too great in ordinary cases. Further, the new motor is different in shaft diameter but not enough different to make it possible for us to key a bushing into the hub of the fan runner.

In the case of belted fans, small changes in speed may be made quickly and easily by merely substituting a motor pulley of slightly different diameter. Greater changes, requiring a change in motor horsepower beyond the motor capacity, or outside the recommended power factor and efficiency range of the motor originally installed, can be easily made by a change to the next size of motor with a suitable pulley.

Belted fans have the great advantage that they permit application of the used motors that are standard as to speed and horsepower for the plant in which they are installed. In case of motor trouble, spares are available. In case of considerable change in speed, involving a change in motor horsepower, the displaced motor will be of shop standard speed and will therefore soon go into service elsewhere, while the new motor may be of standard speed and will often be available within the shop without delay. In case of motor failure in a small shop where few motors are carried in spare stock any available motor of the required capacity or larger will answer to drive the fan temporarily and nothing need be purchased except a motor pulley.

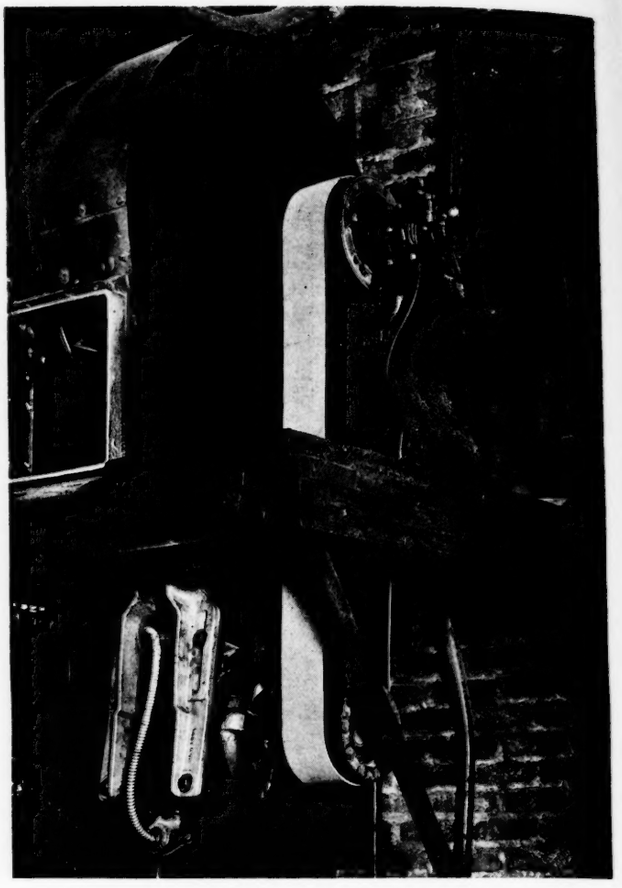
Even when no such motor is available a smaller motor will serve to drive the fan at slightly reduced speed and volume. When the largest available motor has only half the desired horsepower, the fan speed

and volume need be reduced only 20%. Ordinarily such emergency operation will avoid shutting down the process for a few days until the motor can be rewound.

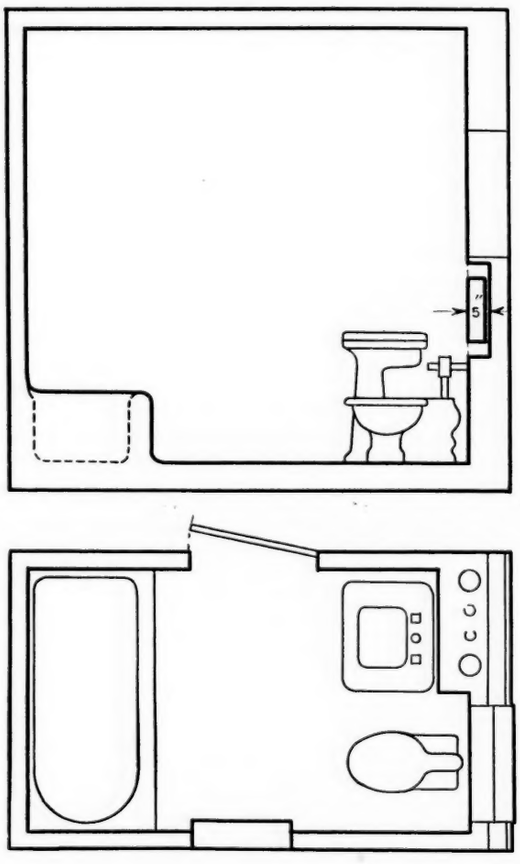
Frequently the necessity for increased capacity, discontinued process, rearrangement of machines served, or some such trouble retires a fan from service. In such cases, in a large industrial plant the opportunity to use a belted fan elsewhere will soon occur. Usually the required speed will be quite different, and the service and capacity different. With a belted fan this simply means putting it into service with another size of pulley and with a motor of different capacity but having a speed conforming to the standard for that capacity in that shop. It is likely to be many years before the same combination of frame, size of fan and speed is needed again.

In years past many direct connected fans have been installed because space considerations dictated a compact installation and the designer feared that belt tensions would not be adequately maintained on a short center drive. That fear was often well founded. At present there are in common use two devices which maintain a correct tension on the slack strand of the belt sufficient to assure a combined belt slip and creep below 1%. Either the pivoted motor base drive or the gravity idler once adjusted will assure the correct belt tension on a short belt drive in spite of any belt stretch which may occur through a period of years. There is no question of the operator's judgment in tightening up on the motor base screws; the tension device once adjusted sets the tension and maintains it substantially constant.

Thirty years ago many fans were built with pulleys of inadequate size for operation at the higher speeds. There is no such difficulty now with fans for exhaust and ventilating work. Almost all multiblade fans are furnished with pulleys designed to transmit the power necessary to drive them, at speeds sufficient to develop at least 6 oz., while paddle fans have pulleys designed for still higher pressure limits. At any pressure or suction which is likely to be met in ventilating work the pulley sizes are ample.



An overhead exhaust fan with flat belt drive



Fitting a Radiator into a Small Bathroom

THE problem of placing the radiator in the small bathrooms of a large apartment house was solved by one firm of consulting engineers by locating an extended surface copper unit in the wall above the closet bowl. The required amount of radiation, as in most apartment bathrooms, was very small, so that the radiator is only a few square feet.

The disadvantage of the location is that the bottom grille of the radiator "chimney" is some distance above the floor. In the case of such small rooms, however, it is doubtful if this disadvantage is of any importance.

The recess, which is 5 in. deep and accommodates a 4 in. wide radiator, has no insulation in back, and the whole installation was quite simple and inexpensive.

Why Not Hot Water?

By J. H. WALKER†

THE trend of residence heating system design for several years has been toward the steam system and away from hot water. Is this an evidence of real superiority, or is it merely another indication that engineering, also, has its fashions?

Hot water heating, as formerly installed, had certain inherent disadvantages, but we seem to have lost sight of the fact that recent developments have almost completely removed those objections. Let us briefly analyze them and examine the actual performance of a residence system of modern design.

Treatises and text-books usually state that the disadvantages of hot water heating are that (1) it requires a larger amount of radiator surface than a steam system; (2) that it is more difficult to design correctly, and (3) that it is not readily responsive to changes in heating demand.

It is true that when designed for a water temperature of, say, 180° a hot water system requires a larger amount of radiator surface, and the piping is usually somewhat more costly than the piping of a steam system. Offsetting this is the elimination of the radiator trap and of certain devices usually regarded as essential to the vapor system. But if one chooses to install a type of hot water system which operates at pressures well above atmosphere and with the boiling point correspondingly raised, the amount of radiator surface need be no greater than for a steam system.

Parenthetically, has anyone ever tried the simple expedient of raising the boiling temperature of the water by the admixture of another fluid? Probably a number of compounds—ethylene glycol being one—could be found which would have the desired effect and would not be too expensive. If a million motor car owners can more or less successfully maintain a solution of prescribed density in their radiators, it ought not to be difficult in a building heating system.

As to the difficulty of designing a system correctly, there was formerly a real dearth of accurate data. In England and in Europe designers were adept at calculating pipe sizes, but in America knowledge of friction drops was either very scanty or those who had it did not disseminate it. Now, however, we have the complete work of Giesecke, in usable form, and based on actual tests on American pipe sizes. The simplicity of the method of design worked out by Professor Giesecke[1] makes the design of the piping system a relatively easy job.

Control of the Heat Supply

Users of steam systems have always been confronted with the fact that the steam heating system is fundamentally very difficult to control. The temperature of steam being constant (or variable through only a limited range), the heat output is difficult to graduate according to the heating requirements. The early steam heating systems were frankly intended to operate intermittently. If a room became too warm, as it frequently did, you turned off the radiator, and when it became cold you turned it on again, or a thermostat did it for you—not always successfully. Then the modulating system came. And now we have variable vacuum and orifice systems—all designed for the job of graduating the heat output of the steam radiator but not perfectly adaptable to a residence system.

Lack of proper temperature control is, undoubtedly, the greatest source of heating discomfort, and *overheating* is one of the worst conditions. In the older hot water systems the evils of overheating were even worse than with steam, but overheating in a gas or oil-fired system can now be almost entirely eliminated by proper control of

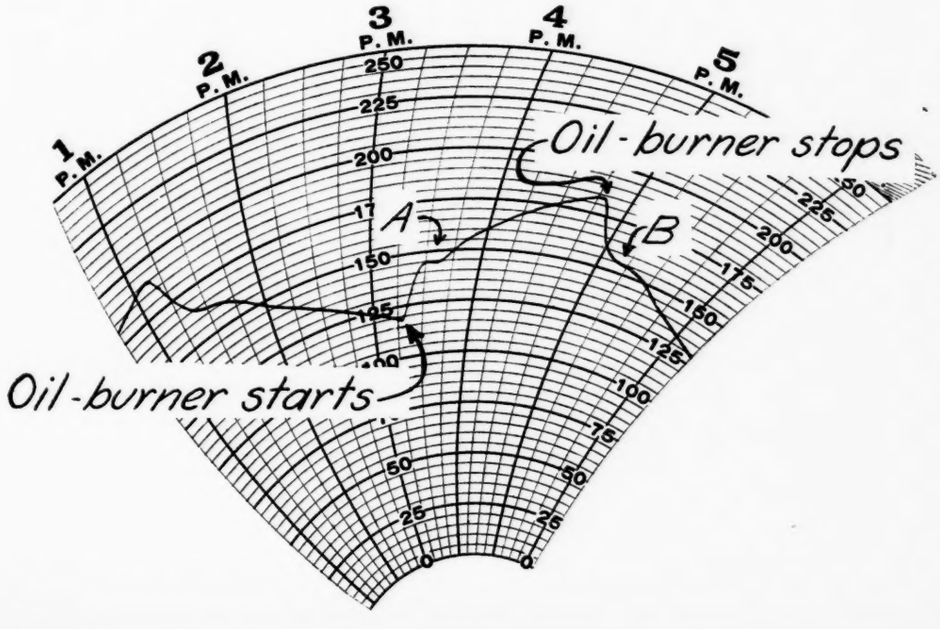


Fig. 1. Temperature cycle in supply main.

†Superintendent of Central Heating, Detroit Edison Company.

[1] "The Design of Gravity Circulation Water Heating Systems," by F. E. Giesecke.

the water temperatures. This is done by the simple expedient of adjusting the maximum water temperature according to the weather conditions, supplemented by a properly located room thermostat. Thus, the old objection that the hot water system is not easily controlled and is unresponsive to changes in the demand for heat is a decidedly out-of-date objection.

Performance of a Hot Water System

Figs. 2 and 3 illustrate the actual performance of an oil-fired hot water system in a residence when subjected to the brutal and revealing test of recording thermometers. If it is ever desired to destroy illusions and to scotch extravagant claims as to the effectiveness of temperature control devices, the simple but rarely used expedient of installing a recording thermometer is to be recommended.

Fig. 2 shows the conditions which existed on a rather cold day, and Fig. 3 on a mild day. The thermostat in this installation is equipped with a clock, and a lower night temperature is carried. The boiler has two controls on the water temperature—an upper temperature limit and a lower temperature limit, the latter being experimentally installed to avoid the occasional formation of condensation when the oil burner starts after a long shut-down period.

Note the cycles of operation of the oil burner—the rather lengthy period of rapidly intermittent operation in the morning as the upper temperature limit comes into action; the rather infrequent periods of operating during the remainder of the day; the rapid cooling of the water after the oil burner cuts off, and particularly the close temperature regulation—plus and minus approximately 1°—not only in the living-room, where the thermostat is located, but in a remote bedroom. The only indications of overheating, barring differences of opinion as to suitable thermostat settings, have been when the sun has shone through unshaded windows. The very satisfactory temperature control is dependent upon the upper limit temperature control which is reset occasionally according to the outside temperature. In early fall and late spring, a water temperature of 120° is most suitable, and in the coldest weather of the past winter the highest temperature carried was about 190° at the boiler, a readjustment once or twice each week being required as weather conditions change. Automatic controls are now obtainable which will continuously regulate this upper water temperature limit according to the prevailing outdoor temperatures, thus eliminating the need of hand adjustments. Proper control of the *water temperature* is the key to satisfactory heating with a hot water system.

The uniformity of temperature in different rooms has

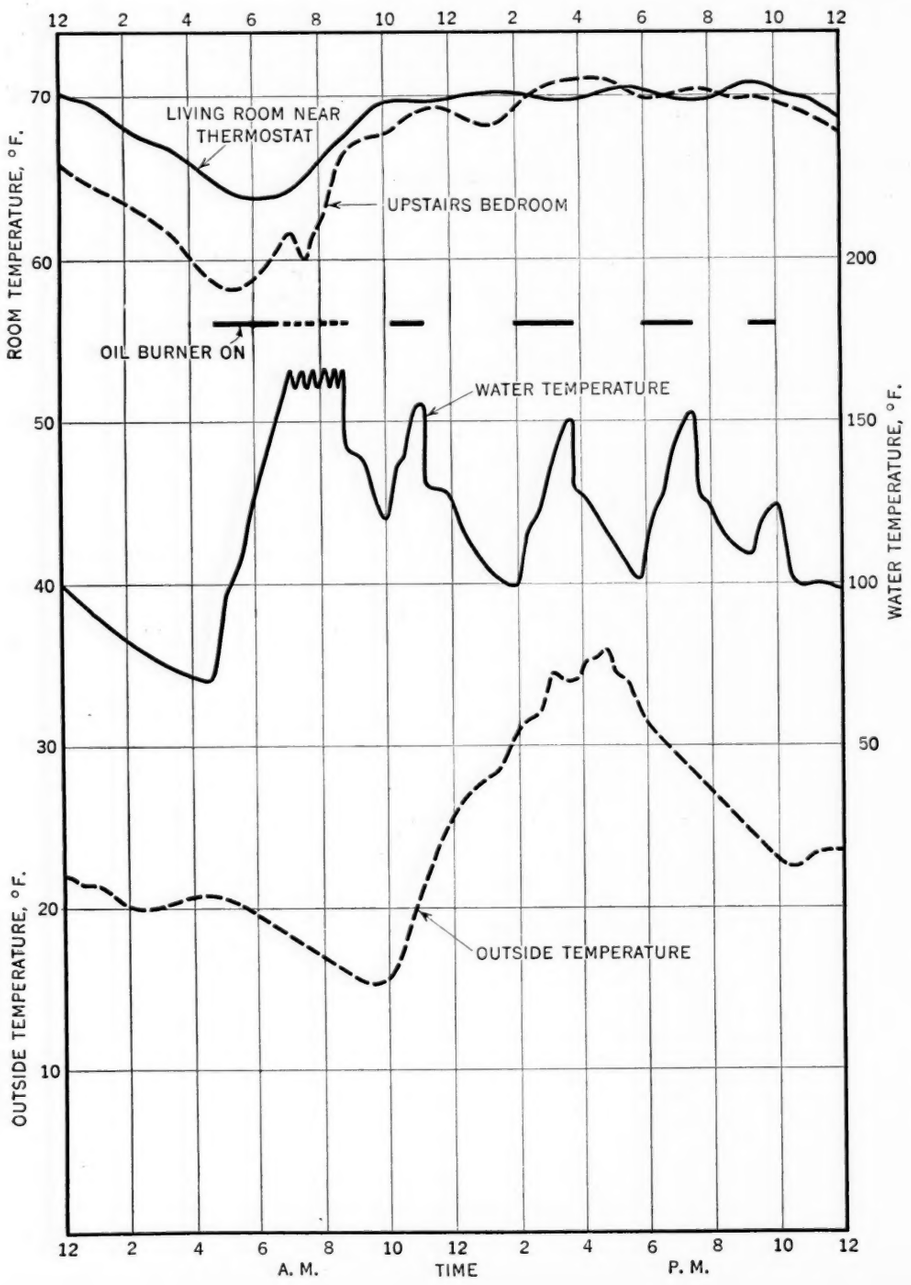


Fig. 2. Operation of heating system on a cold day.

been quite noticeable. In many steam heated houses there is a sharp rise in room temperature in all of the rooms, when the thermostat calls for heat, with an appreciable overrun. During the ensuing off period, the temperatures are falling and the rooms having the more rapid *rates of cooling* reach uncomfortably low temperatures before the key room cools to the opening point of the thermostat. With the hot water system the temperature rise and temperature drop in the rooms are slower and the ranges of temperature appreciably less. It is a fact often lost sight of that with either the steam or water system under the on-and-off thermostatic control the heating effect in the rooms is intermittent rather than constant, but with hot water it is on the whole much less variable than with steam.

Although the control devices referred to might be regarded as rather complex, they are merely evidences of the trend to more elaborate and complete controls which appear to be unavoidable if we are to obtain the greatly needed advances in heating comfort. With the automobile as an outstanding example of complicated devices rendered dependable by improvements in design and in servicing, we need not be disturbed by a similar line of progress in heating.

Critical Points in the Circulation Cycle

An interesting phenomenon was observed in the temperature charts of the water in one of the supply mains, as shown in Fig. 1. When the oil burner starts, the water temperature in the main rises rapidly for a few minutes as a slow circulation begins. Presently, however, there occurs the flat spot, A, in the temperature curve followed by a further rise. The explanation of this is that a critical point has been reached at which the circulation through the system suddenly becomes quite rapid and the cool water from the radiators rushes back to the boiler in large volumes, temporarily checking the temperature rise.

A somewhat similar flat spot, B, appears in the cooling curve. This is due to a sudden reduction in the rate of flow. These phenomena were discovered, or at least first explained, several years ago by A. H. Barker of England, who worked out the rather complicated mathematics involved. They have no great practical significance but serve to illustrate the action of the varying circulating forces in a system operating in cycles.

One fault will be found with the performance, as shown in Figs. 2 and 3—the house is rather slow in warming up in the morning. The remedy would be to set the thermostat for an earlier hour, were a 70° temperature at the rising felt to be important. A control could undoubtedly be developed to give a high water temperature during the warming-up period but, lacking

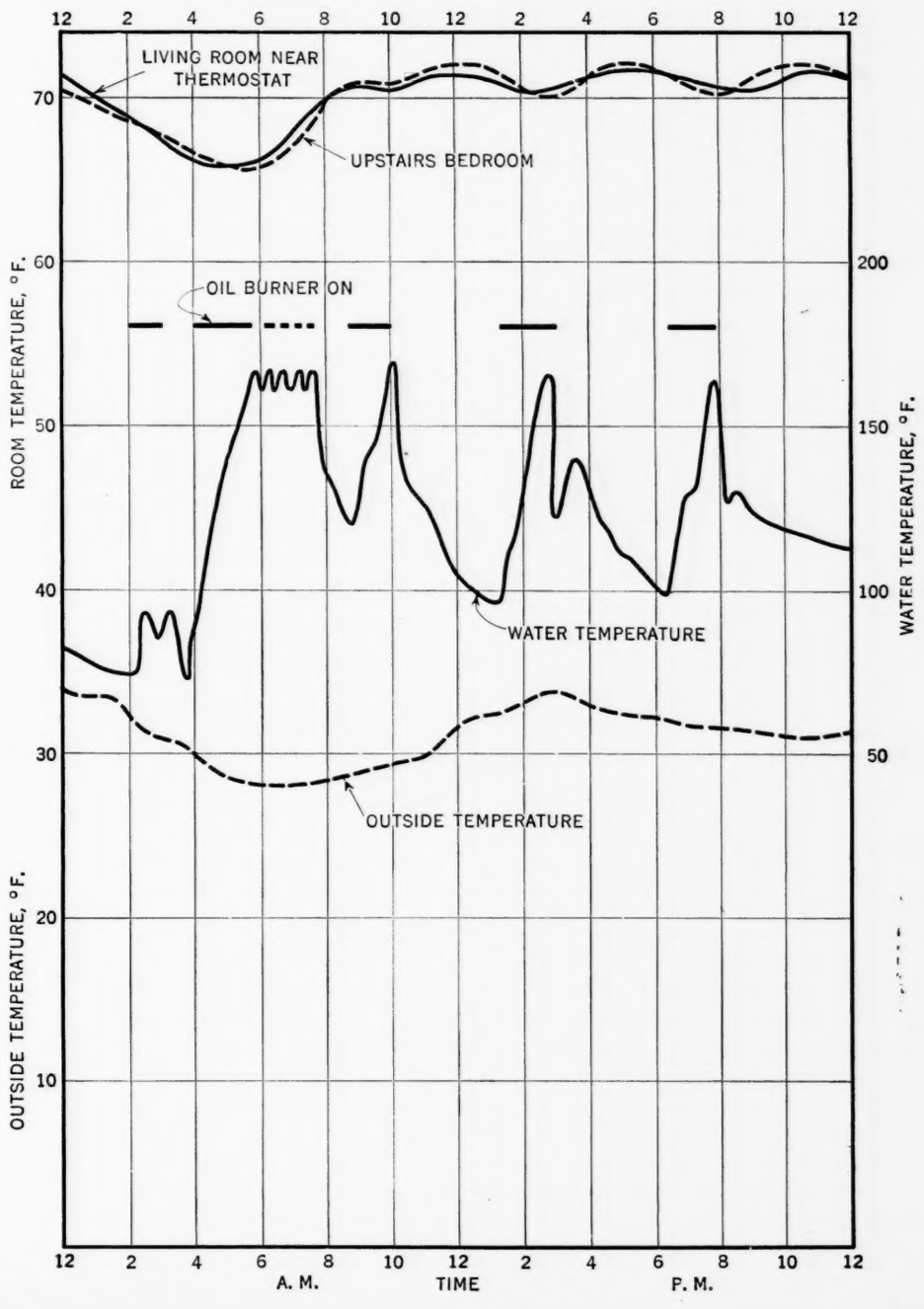


Fig. 3. Operation of heating system on a mild day.

it, one must accept in this method of hot water heating a rather long warming-up period.

This experience has demonstrated, at least to the satisfaction of the writer, that the hot water system, with an oil burner and with a properly handled water temperature control, gives highly satisfactory results as to even and uniform temperatures and freedom from overheating which are seldom equalled with a steam system.

Another advantage of the hot water system is freedom from corrosion, which is sometimes encountered in steam systems. The hot water system is full of degasified and chemically inert water year after year.

Calculation of Pipe Sizes

The pipe sizes in this system, which appear to be adequate in every respect, were carefully worked out by the Giesecke method. It was interesting to note that the tables of pipe sizes, developed from experience by the late John R. Allen, gave pipe sizes which checked almost exactly with the calculated sizes. The only variation was in the case of some of the radiator branches which worked out as ½ in. pipes but which the table gave as ¾ in. As a matter of interest the tables are reproduced here.[1]

[1] From "HEATING AND VENTILATION," Allen and Walker, second edition.

TABLE 1

SIZE OF MAINS

Assumed Length 100 Ft., Temperature Drop in Radiators 20°

Pipe diam.	Capacity, square feet of direct radiation		
	Two-pipe upfeed	One-pipe upfeed	Overhead
1¼	75	45	130
1½	110	65	190
2	200	121	340
2½	310	190	530
3	540	330	920
3½	780	470	1,330
4	1,100	650	1,800
5	1,900	1,100	3,200
6	3,000	1,800	5,000
7	4,300	2,700	7,200
8	5,900	3,500	9,900

TABLE 2

SIZE OF RISERS

Assumed Temperature Drop in Radiators, 20°

Pipe size	Upfeed				Downfeed risers, not exceeding four floors
	First floor	Second floor	Third floor	Fourth floor	
1	33	46	57	64	48
1¼	71	104	124	142	112
1½	100	140	175	200	160
2	187	262	325	375	300
2½	292	410	492	580	471
3	500	755	875	1,000	810

Panel Heating Tests Being Conducted in Great Britain

EXTENSIVE investigations into the science of heating, ventilating, and air conditioning are being conducted by the building research board of the Department of Scientific Research of Great Britain, as a result of numerous inquiries pertaining to the subject. Close collaboration is developing between the research board and the Institution of Heating and Ventilating Engineers in regard to work on these problems, the board announced in its annual report, recently issued.

Panel heating systems have been tested in an experimental house which was designed to simulate, as closely as practicable, the conditions existing in one of a pair of semi-detached houses. For comparison of this heating system with convection heating, the rooms of the house have been maintained day and night at a temperature comfortable for sedentary occupation. From a graph plotted from experimental data, it is apparent that the heat requirements of rooms warmed without ventilation, by the panel system, are approximately 14% greater than with convective heating. It is said, however, that the relative advantage of the two methods is not likely to remain the same with freely ventilated rooms, and future research may prove it to be reversed.

Heating by low temperature radiation from panels has become increasingly popular, but there are a number of difficulties from the constructional standpoint that must be overcome, especially in cases where heating is carried out by water. Tests were carried out on the efficiency of furnace slab coverings for panel heating coils, in comparison with similar coils of like dimensions with ordinary plaster covering. It was found that the heat output of the furnace panel was considerably less than that of the plaster panel. A small scale test, with various fillings between the heating coils and the slabs, revealed that results may be improved by a filling of aluminum filings.

Effect of pipe diameter and spacing on the heat output of ceiling panels was also investigated. It was demonstrated that the heat emission of a ¾ in. pipe was not materially greater than that of a ½ in. pipe, and that wider spacing of the pipe may be economically advantageous, as a large reduction in pipe length can be made at the expense of a small increase of panel area.

Heat transmission through roofs, especially those of the type where thin sheet materials are used, has been studied, and an air conditioning plant is being installed to provide data on this subject.

Calculating Radiation by Use of Tables

By W. F. FISCHER

IT is quite generally admitted that the so-called rule-of-thumb methods which were so frequently used in the past for estimating the amount of radiation required to offset the heat losses from a given room or building were, to say the least, mere approximations only, which gave results far from satisfactory in the majority of cases.

Some of these old rules are still in use and still appear in print from time to time, but it is far better for the heating contractor or estimator to make up his own short rules for estimating radiation, based on the B.t.u. method, than to adopt some rule which may give fairly accurate results in some cases and far from satisfactory results in others.

A short rule for estimating radiation based on an outdoor temperature of 15° F. and an indoor temperature of say 65°, would hardly be expected to apply to other localities where the outdoor temperature is zero or lower, and to rooms which are to be heated to temperatures other than 65°. Then, again, a short rule applicable to a given type of building construction, such as wooden-frame dwellings, cannot be applied to other types of building construction with any degree of accuracy unless modified by certain conversion factors.

In this age of engineering research we have available reliable data pertaining to the heat loss through walls of almost every description used in modern building construction, so that there is no longer any excuse for the use of any of the obsolete rule-of-thumb methods, developed at a time when such data were meager, or not available at all.

It is the purpose of the present article to describe a method developed by the writer, and used for many years in his own practice, for simplifying the calculations necessary to determine the size of radiator, or radiators, required to offset the heat loss from a building. The method is based on use of the following:

In residential buildings, the wall height from floor line to ceiling is in the neighborhood of 9 ft.

In and around the vicinity of New York, it is customary to assume 0° as the minimum outdoor temperature and 70° as the indoor temperature, which gives a temperature difference of 70°.

Each square foot of direct cast iron steam radiation is assumed to give off at least 240 B.t.u. per hr.

Using these figures, the number of square feet or

TABLE 1

SQ. FT. OF DIRECT CAST IRON (STEAM) RADIATION, R, REQUIRED PER LIN. FT. OF EXPOSED WALL 9 FT. HIGH, FOR A TEMPERATURE DIFFERENCE OF 70°

Type of Wall	Outside Finish	Furred, Lath and Plaster Inside			Inside Surface Plain, No Interior Finish		
		U	R	Glass Constant G	U	R	Glass Constant G
12″ Hollow tile	Stucco	0.144	0.378	0.288	0.201	0.528	0.271
8″ Hollow tile	Stucco	0.184	0.483	0.276	0.291	0.765	0.245
6″ Hollow tile	Stucco	0.196	0.515	0.273	0.320	0.840	0.237
12″ Hollow tile	4″ Brick veneer	0.130	0.342	0.292	0.175	0.460	0.279
8″ Hollow tile	4″ Brick veneer	0.162	0.425	0.282	0.240	0.630	0.260
6″ Hollow tile	4″ Brick veneer	0.171	0.450	0.280	0.260	0.683	0.254
12″ Concrete	None	0.226	0.594	0.264	0.411	1.080	0.210
8″ Concrete	None	0.254	0.668	0.256	0.512	1.345	0.180
6″ Concrete	None	0.270	0.710	0.251	0.583	1.532	0.160
18″ Brick	None	0.152	0.400	0.286	0.218	0.573	0.266
13″ Brick	None	0.179	0.470	0.278	0.278	0.731	0.249
9″ Brick	None	0.209	0.550	0.269	0.358	0.942	0.226
12″ Concr. block	None	0.183	0.482	0.277	0.287	0.755	0.246
10″ Concr. block	None	0.198	0.520	0.272	0.327	0.861	0.235
8″ Concr. block	None	0.215	0.565	0.267	0.377	0.992	0.220
Wood frame, sheathing	4″ Brick veneer	0.216	0.568	0.266			
Wood frame, sheathing	Clapboards	0.227	0.597	0.264			
Wood frame, sheathing	Wood shingles	0.227	0.597	0.264			
Wood frame, sheathing	Stucco	0.257	0.675	0.255			

fractions of a square foot of direct cast iron steam radiation, R, required for each lineal foot (not square foot) of wall, 9 ft. high, exposed to the weather, may be estimated as follows:

$$R = \frac{9 \times 70}{240} U = 2.625U \qquad (1)$$

where R is the square feet of steam radiation per lineal foot of exposed wall, and U is the coefficient of heat transmission in B.t.u. per hour per square foot of wall surface per 1° temperature difference.

The values of U are the coefficients given in standard reference works. A few examples are given in the third and sixth columns of Table 1 for various types of wall construction. The coefficients are based on a 15 m.p.h. wind against the wall.

The values of R, as computed from (1) are given in the fourth and seventh columns of Table 1.

As an example, the coefficient, U, for an 8 in. hollow tile wall with 4 in. brick veneer on the outside and plaster on lath over furring strips on the inside (leaving an air space between the tile and the plaster) is given in the third column of the table as 0.162. Multiplying this by $\left(\frac{9 \times 70}{240}\right)$ or 2.625, we find that R = 0.425 sq. ft. of direct cast iron steam radiation will be required for each lineal foot of exposed wall, 9 ft. high, as given in the fourth column of Table 1.

The transmission coefficient for single glass windows is 1.13. The difference between the glass coefficient, G, and the wall coefficient, U, as given in the table, will then be (1.13 — U). Multiplying this by 70, the difference between the inside and outside temperatures, and dividing by 240, we have

$$G = \frac{70\ (1.13 - U)}{240} = 0.292\ (1.13 - U) \qquad (2)$$

where G is the glass constant for the particular type of wall under consideration. It varies with different types of wall construction.

Values of G for each type of wall construction considered here are given in the fifth and eighth columns of Table 1.

If we figure any particular type of wall as being solid throughout, without any window openings, and multiply the square feet of glass surface by G, it is evident that we obtain the same results as we would by deducting the window openings and then estimating the actual heat loss through the glass in the regular way.

To show how G was calculated, for an 8 in. hollow tile brick veneer wall, lath and plaster inside on furring strips, 70° difference in temperature, the heat loss per square foot per hour is

$$0.162 \times 70 \times 1 = 11.34 \text{ B.t.u. per hr.}$$

Dividing by 240, we have 0.0473 sq. ft. of steam radiation required per square foot of wall.

The heat loss per hour through 1 sq. ft. of single glass is 1.13 × 70 = 79.10 B.t.u., or 0.329 sq. ft. of steam radiation.

The difference in radiation between that required for the glass and that required for the wall is

$$0.329 - 0.0473 = 0.282 \text{ sq. ft.}$$

This is the G as given in the fifth column of Table 1.

The following example will serve to illustrate the use of Table 1 in estimating the radiation required to offset heat losses through an exposed wall and glass surface in the same wall:

A room 14 ft. long, 12 ft. wide and 9 ft. high has two sides exposed. The inside temperature is 70°, outside temperature zero; walls are 9 in. brick, furred and plastered inside; there are 40 sq. ft. of glass surface in the walls. How many square feet of direct cast iron steam radiation must be provided?

TABLE 2

SQ. FT. OF DIRECT CAST IRON (STEAM) RADIATION, F, REQUIRED PER SQ. FT. OF EXPOSED FLOOR OR CEILING AREA FOR A TEMPERATURE DIFFERENCE OF 15° AND 35°

Type of Floor	No Ceiling Below			Lath and Plaster Ceiling		
	U	F 15° Diff.	F 35° Diff.	U	F 15° Diff.	F 35° Diff.
1″ Wood, on wood joists	0.440	0.0275	0.0643	0.234	0.0147	0.0342
13/16″ Oak on 1″ wood floor	0.340	0.0213	0.0497	0.210	0.0131	0.0307
4″ Concrete, plain	0.508	0.0318	0.0742	0.264	0.0165	0.0285
6″ Concrete, plain	0.452	0.0283	0.0661	0.248	0.0155	0.0363
8″ Concrete, plain	0.408	0.0255	0.0597	0.234	0.0147	0.0342
1″ Wood on 4″ concrete	0.364	0.0228	0.0532	0.219	0.0137	0.0320
1″ Wood on 6″ concrete	0.334	0.0209	0.0488	0.208	0.0130	0.0304
1″ Wood on 8″ concrete	0.310	0.0194	0.0453	0.198	0.0124	0.0290
Lath and plaster ceiling, no floor above				0.502	0.0344	0.0604

Type of Floor (on Ground)	Type of Upper Floor Finish					
	Plain Concrete		1″ Y. P. Flooring on Wood Sleepers		1″ Terrazo or Tile, on Cement	
	U	15° F.	U	15° F.	U	15° F.
4″ Concrete on cinders	0.556	0.0348	0.388	0.0237	0.526	0.0329
6″ Concrete on cinders	0.490	0.0306	0.355	0.0222	0.467	0.0292
8″ Concrete on cinders	0.439	0.0275	0.327	0.0204	0.420	0.0263

We have 14 + 12 = 26 lin. ft. of exposed wall (9 ft. high). From Table 1 we find for a 9 in. brick, furred and plastered wall, the values, R = 0.55 and G = 0.269, respectively. Hence, there will be required

$$(26 \times 0.55) + (40 \times 0.269) = 25 \text{ sq. ft.}$$

of steam radiation or 6000 B.t.u. per hr. for the walls and glass.

Ceiling Heights Other Than 9 ft.

If Table 1 is to be used for ceiling heights other than 9 ft., we must multiply the values of R as given in the table by the actual ceiling height in feet, and divide the product thus obtained by 9. For example, for a ceiling height of 8 ft. 4 in. (approximately 8.33 ft.), in the above example instead of 9 ft., we should have for a new value of R for this particular wall,

$$\frac{0.55 \times 8.33}{9} = 0.511, \text{ approximately.}$$

The radiation in the above example would then be $(26 \times 0.511) + (40 \times 0.269) = 24$ sq. ft. approximately, instead of 25 sq. ft. as figured for a 9 ft. ceiling.

For a wall 11 ft. high instead of 9 ft., the value of R would increase to

$$\frac{11 \times 0.55}{9} = 0.672, \text{ approximately.}$$

Floors and Ceilings

When estimating the heat loss through floors to unheated rooms or space below, and through ceilings to unheated attic spaces above, it is usually sufficiently accurate to assume the temperature of the unheated spaces as a mean between the inside and outside temperature; that is, if the outside temperature is taken as zero and the inside as 70°, the temperature of the unheated space may be assumed as the mean of these two temperatures, or 35°.

The room temperature, for heated rooms, is usually taken as 70° at the breathing level, about 5 ft. above the floor line. We should not forget, however, that the temperature close to the floor line may be as low as 65° or even lower; while the temperature at a point close to the ceiling may be as high as 80° or even higher.

In this case, then, a heated room on the second floor of a building, located directly above a similar heated room on the first floor, would receive heat through the floor from the heated room below, due to this higher ceiling line temperature.

Assume the ceiling line temperature as 80° and the floor line temperature overhead as 65°, a difference of 15°; the coefficient, U, for an ordinary wood floor on joists, with lath and plaster ceiling below is 0.55 B.t.u. per hr. per sq. ft. per degree difference in temperature, or $0.55 \times 15 = 8.25$ B.t.u. for a 15° difference. The total heat loss through the floor of a room 12 ft. x 14 ft. would then be at the rate of $12 \times 14 \times 8.25 =$ 1,386 B.t.u. per hr., or the equivalent of 5.8 sq. ft. of direct cast iron steam radiation, approximately.

This means, then, that the radiation in the heated room below should, practically speaking, be increased by 5.8 sq. ft. to offset the heat lost through the ceiling, and the radiation in the room above decreased by a like amount.

For floors laid directly on the ground, with no air space beneath them, the temperature of the ground is usually taken as 50° in zero weather. If the temperature above the floor is taken as 65° we have also a temperature difference of 15° to consider when estimating the heat lost through such floors.

Let F be the square feet of direct cast iron steam radiation required to offset the heat loss per hour through 1 sq. ft. of floor or ceiling construction. Then, for a temperature difference of 15°, we have

$$F \text{ (15° difference)} = \frac{15}{240}(U) = 0.0625\ U \qquad (3)$$

and for a temperature difference of, say 35°, we have

$$F \text{ (35° difference)} = \frac{35}{240}(U) = 0.146\ U, \text{ approx.} \qquad (4)$$

where U is the coefficient of transmission for the floor or ceiling.

TABLE 3

SQ. FT. OF DIRECT CAST IRON (STEAM) RADIATION, C, REQUIRED PER SQ. FT. OF ROOF AREA FOR A TEMPERATURE DIFFERENCE OF 70°

Type of Roof (Flat)	No Ceiling Below		Plaster Ceiling on Lath		Hung Ceiling on Metal Lath	
	U	C 70° Diff.	U	C 70° Diff.	U	C 70° Diff.
2″ Concrete composition roofing	0.658	0.192			0.300	0.088
4″ Concrete composition roofing	0.568	0.166			0.280	0.082
6″ Concrete composition roofing	0.500	0.146			0.262	0.077
Flat metal, composition roofing	0.781	0.228			0.324	0.095
Corrugated iron, plain	1.500	0.438				
1″ Wood, composition roofing	0.485	0.142	0.258	0.076		
1½″ Wood, composition roofing	0.386	0.113	0.277	0.081		
2″ Wood, composition roofing	0.345	0.100	0.213	0.063		
3″ Wood, composition roofing	0.256	0.075	0.175	0.051		
Pitched Roofs Over Heated Attics						
Wood shingles on wood strips	0.483	0.141	0.246	0.072		
Asphalt shingles on sheathing	0.518	0.152	0.259	0.076		
Composition roofing on sheathing	0.518	0.152	0.259	0.076		
Slate or tile on sheathing	0.549	0.160	0.262	0.077		

TABLE 4

SQ. FT. OF DIRECT CAST IRON (STEAM) RADIATION, A, REQUIRED PER SQ. FT. OF FLOOR AREA, TO OFFSET THE HEAT LOSS IN A ROOM DUE TO AIR LEAKAGE, FOR A TEMPERATURE DIFFERENCE OF 70°

Ceiling Height in Feet	Number of Air Changes Per Hr.				
	1	1½	2	2½	3
	Sq. Ft. of Direct Cast Iron (Steam) Radiation Per Sq. Ft. of Floor Area Values of A				
8.0	0.0420	0.0630	0.0840	0.1050	0.1260
8.5	0.0446	0.0670	0.0893	0.1116	0.1339
9.0	0.0473	0.0709	0.0945	0.1182	0.1418
9.5	0.0490	0.0735	0.0980	0.1225	0.1470
10.0	0.0525	0.0788	0.1050	0.1313	0.1575
10.5	0.0550	0.0825	0.1100	0.1375	0.1650
11.0	0.0578	0.0866	0.1155	0.1443	0.1733
11.5	0.0605	0.0908	0.1210	0.1513	0.1815
12.0	0.0630	0.0945	0.1260	0.1575	0.1890

The values of F are given in Table 2 for a temperature difference of 15°, and also for a temperature difference of 35°. Sleeping rooms on the second floor or third floor of a dwelling are sometimes figured for a temperature of 60° instead of 70°. In such cases, the ceiling temperature may be about 70° and the attic space 35°, a difference of 35°. For a temperature difference of 45° the values of F for a 15° difference, may be multiplied by 3.

Roofs

Let C be the square feet of direct cast iron radiation required to offset the heat lost in one hour through each square foot of roof surface directly above a heated room.

For a temperature difference of 70° we have

$$C = \frac{70}{240}(U) = 0.292\ U, \text{ approx.} \qquad (5)$$

where U is the coefficient of transmission for the roof construction.

Values of C, as computed by the above formula, are given in Table 3 for various types of roof construction.

For a ceiling temperatue of 80° and outside air at zero, the values of C, as given in the table, should be multiplied by the factor $\frac{80}{70}$ (= 1.143); or increased by, say, 15%.

Air Change

Although the most accurate method of estimating the heat loss from a given room due to air leakage is that based on the infiltration of air per lineal foot of window crack, it is accurate enough to assume a certain number of air changes per hour, depending on the exposure of the room and the tightness of the windows or doors opening to the outside air.

For windows and doors of good construction and reasonably tight, not necessarily weatherstripped, the following air changes are recommended as a basis of computing the heat loss due to air leakage into the room from the outside:

Rooms having 1 side exposed,	1 air change per hr.
Rooms having 2 sides exposed,	1½ air changes per hr.
Rooms having 3 sides exposed,	2 air changes per hr.
Rooms having 4 sides exposed,	2 air changes per hr.
Entrance halls	2 to 3 air changes per hr.
Reception halls	2 air changes per hr.

The figures given above are exclusive of any additional air required for ventilation.

With a ceiling height of 9 ft., there will be 9 cu. ft. of air per sq. ft. of floor surface. Taking the outside air at zero and the room temperature as 70°, the density of air at 70° as 0.075 lb. per cu. ft., and the specific heat of air as 0.24, we may write the following formula:

$$A\text{(for 1 air change per hr.)} = \frac{9 \times 0.075 \times 0.24 \times 70}{240} \qquad (6)$$

or A = 0.04725 sq. ft. of direct cast iron steam radiation per square foot of floor area for one air change per hour for a temperature difference of 70° and a ceiling height 9 ft.

The values of A, for different ceiling heights and several air changes per hour, are given in Table 4, for a temperature difference of 70°.

Tables 1 to 4, inclusive, may be used to estimate the amount of direct radiation required in all cases where the temperature difference is 70°. As an example showing their use, let us assume a room to have:

(a) Walls of wood-frame construction, wood shingles on sheathing boards on 4 in. studs, plastered inside on wood lath;

(b) 13/16 in. hardwood floors laid on top of ⅞ in. boards on wood joists, no ceiling below, and space below floor unheated;

(c) Ordinary lath and plaster ceiling on joists, with ⅞ in. flooring boards above, attic space unheated;

(d) 1½ air changes per hr., difference in temperature 70°, ceiling height 9 ft.

(a) From the figure we have 26 lin. ft. of exposed wall and 40 sq. ft. of glass. From Table 1, fourth and fifth columns, second row from bottom, we find for this wall the values R = 0.597, G = 0.264. Hence, there will be required for the combined wall and glass surface,

$(0.597 \times 26) + (0.264 \times 40) = 26.0$ sq. ft. of steam radiation, approx.

(b) Taking the temperature of the unheated space beneath the floor as 35° and the temperature above the floor as 65°, we have a difference of 30°. For this floor (no ceiling below) we find from Table 2, third column, second row from the top, a value of F for a 15° difference, equal to 0.0213; for a 30° difference this value increases to $(0.0213 \times 2 =)$ 0.0426. Hence, since the floor area is 168 sq. ft., there will be required $0.0426 \times 168 = 7.2$ sq. ft. (approx.) of radiation to offset the heat loss through the floor to the cold room below.

(c) Taking the temperature close to the ceiling as 80° and the unheated attic space as 35°, we have a temperature difference of 45°. For this ceiling, with ⅞ in. attic floor above, we find from Table 2, sixth column, first line, the value F = 0.0147 for a temperature difference of 15°; for a 45° difference, then, we have $F = 3 \times 0.0147 = 0.0441$. Thus, there will be required $0.0441 \times 168 = 7.4$ sq. ft. of radiation to provide for ceiling heat loss.

(d) The radiation required to offset the heat loss due to 1½ air changes per hr., is found by multiplying the value of A, Table 4, column 3, third line; that is, A = 0.0709, by the floor area, 168 sq. ft., which gives us $0.0709 \times 168 = 11.9$ sq. ft.

Summing up, we have

Wall and glass combined	26.0 sq. ft.
Floor	7.2 sq. ft.
Ceiling	7.4 sq. ft.
Air changes	11.9 sq. ft.
Total	52.5, say, 53 sq. ft.

of direct steam radiation, at 240 B.t.u. per sq. ft. per hr.

Checking this by the conventional method, we have

Glass, $40 \times 1.13 \times 70$	= 3,164 B.t.u.
Wall $(234 - 40) \times 0.227 \times 70$	= 3,083 B.t.u.
Floor, $168 \times 0.34 \times 30$	= 1,714 B.t.u.
Ceiling, $168 \times 0.234 \times 45$	= 1,769 B.t.u.
Air changes $(1512 \times 1½) \times 0.075 \times 0.24 \times 70$	= 2,858 B.t.u.
Total hourly heat loss	= 12,588 B.t.u.

$12{,}588 \div 240 = 52.5$ approx., or 53 sq. ft. of steam radiation.

No account has been taken of the direction in which the walls are exposed and the radiation should be increased accordingly (if the exposure is in a northerly direction) by the usual percentage.

By the use of carefully prepared tables, similar to those illustrated above, much labor can be saved in estimating radiation, and the results are just as accurate as those obtained in the regular way, by the B.t.u. method.

The square feet of steam radiation, as obtained in the above manner, should be increased by at least 10% or 15% for vapor heating, due to the lower pressure and temperature of the steam circulated to the radiators in such systems.

If hot water, instead of steam, is used as the heating medium, the radiation should be increased about 60% to allow for the lower temperature of the water. This is based on a heat emission of 150 B.t.u. per hr. per sq. ft. for hot water, as compared with 240 B.t.u. for steam.

From the information given above, it should be a simple enough matter for anyone to make up a set of tables to suit his own requirements as to outdoor and indoor temperatures, different types of building construction, different ceiling heights, etc., and once thoroughly familiar with their use, much time can be saved, which will more than pay for the trouble of making the tables.

Recent Developments in Heat Transmission

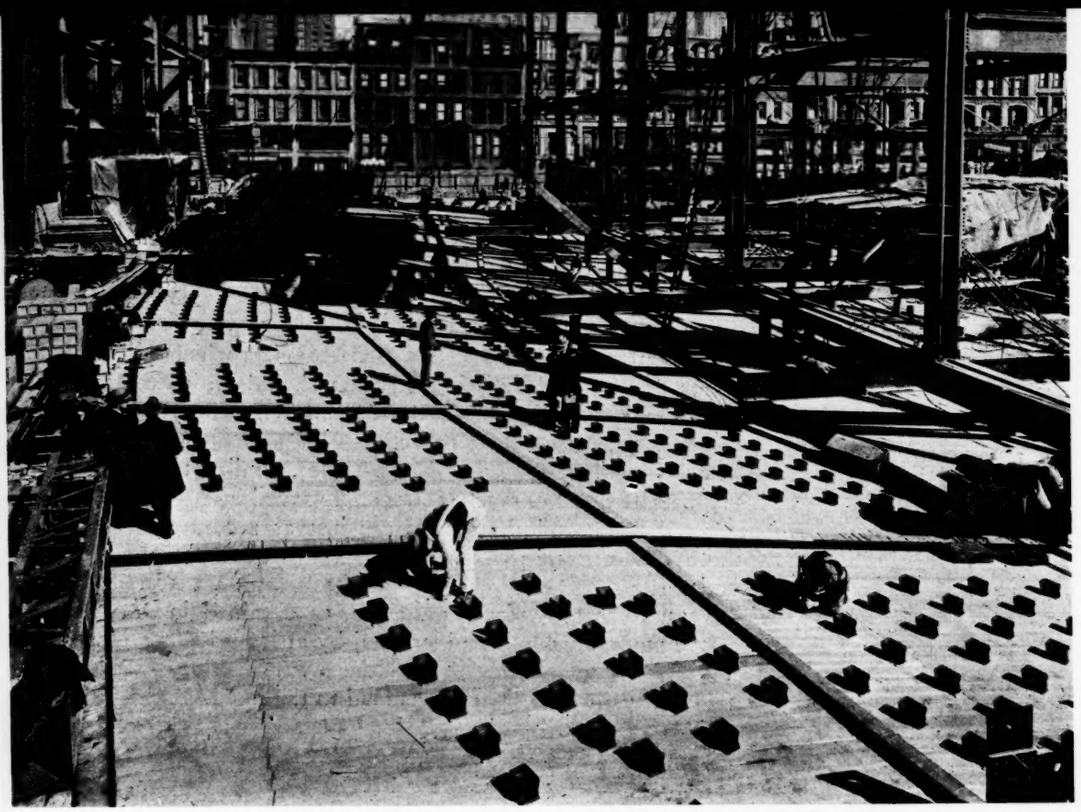
THE mechanism of heat flow in porous or fibrous insulating materials has been discussed in papers by Heilman, Finck, and the writer. Finck shows that a number of dry, fibrous materials, such as jute, bagasse and wood pulp, reach a minimum conductivity (at 90°) of about 0.28 at a density of 5 lb. per cu. ft.

Considerable interest has been manifested lately in the use of multiple air spaces, separated by sheets of aluminum foil or paper, for thermal insulation. Conductivities as low as 0.208 have been reported for aluminum foil with 3 spaces per inch, and 0.27 for paper with 10 spaces per inch.

The conductivities of a number of other materials of importance to the refrigeration industry have been determined more accurately in recent years. Valuable information and data on heat insulators are contained in the papers by Griffiths and by Townshend and Williams. Some new data on water, toluol, SO_2, CO_2 and NH_3 (liquids) are reported by Jakob. Trautz and Zundel have presented a survey of the methods and data for air, hydrogen and CO_2 gas, and Jakob and Erk have carried the measurement of k for ice down to a temperature of —200°. These results are included in Table 1, together with values of k for a few other materials of interest.—*W. J. King, in a paper presented before the spring meeting, American Society of Refrigerating Engineers, June* 9-11, 1932.

TABLE 1.
THERMAL CONDUCTIVITIES OF VARIOUS MATERIALS

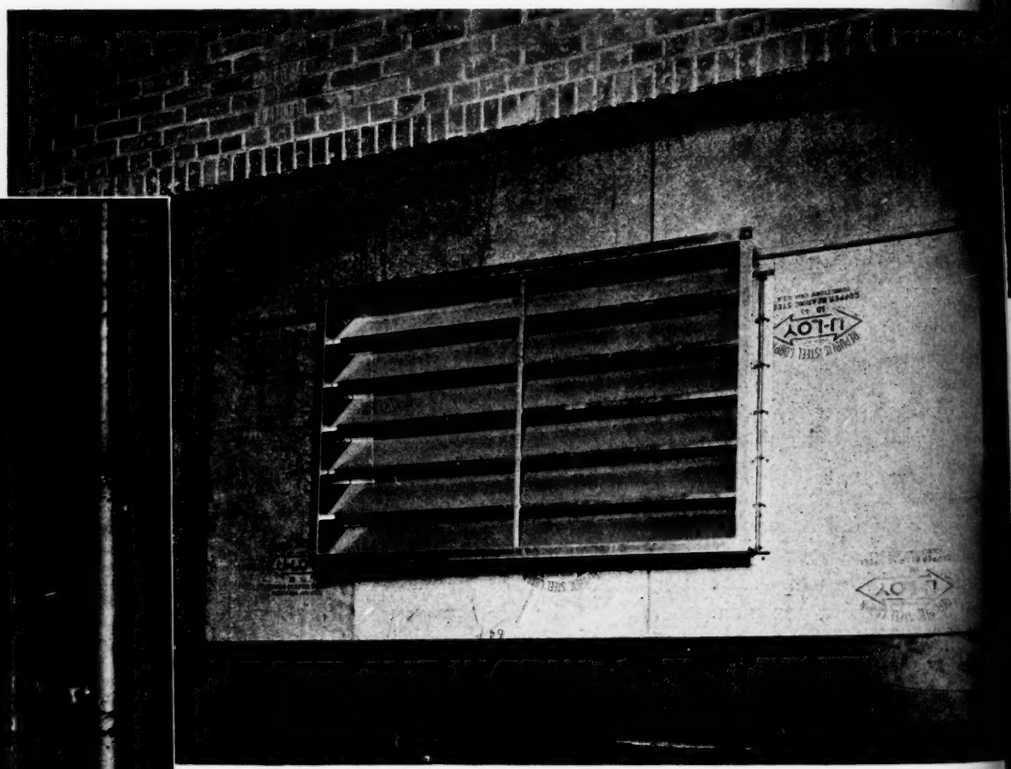
	Temp. F.°	k B.t.u. per hr. per ft.² per° F. per in.
Air	32	0.163
Aluminum	68	1460.0
Aluminum foil, 3 air spaces per in.	68	0.208
Ammonia (liquid)	68	4.03
Ammonia (gas)	32	0.149
CO_2 (liquid)	68	1.45
CO_2 (gas)	32	0.097
Copper	68	2665.0
Cornstalk pulp	90	0.289
Ebonite, cellular expanded	76	0.235
Glass wool	44	0.276
Hydrogen (gas)	32	1.13
Ice	32	15.0
Ice	—200	28.0
Paper, 10 air spaces per in.	50	0.27
Steel	68	300.0
SO_2 (liquid)	68	2.34
SO_2 (gas)	32	0.056
Toluol	68	1.05
Water	68	4.10



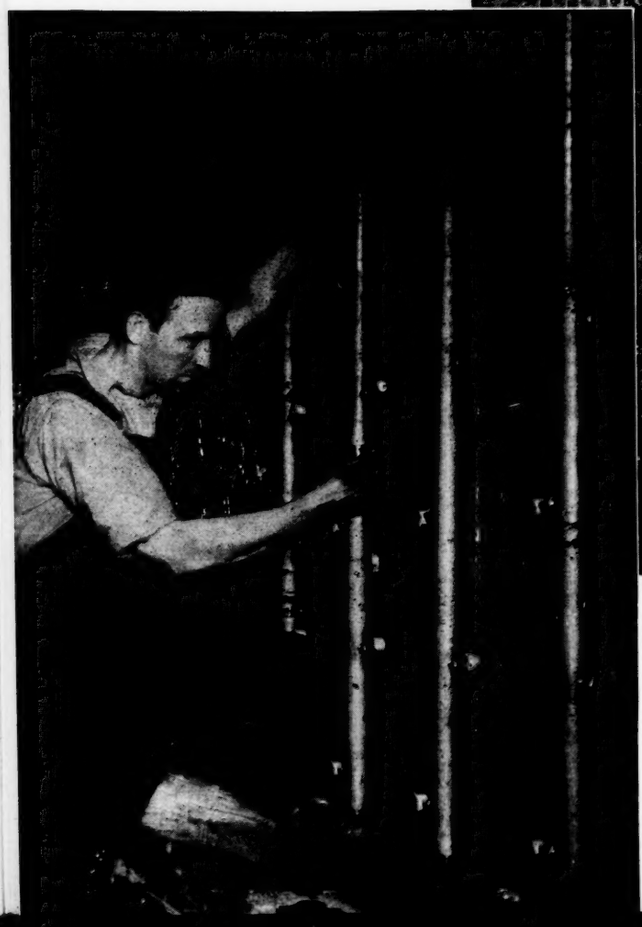
Rockefell Ce

Construction is progressing
New York's $250,000,
pleted Rockefeller Center
building in point of floor
over a mile of shop front
are an important part of
world's largest broadcast

Air will be exhausted from the International Music Hall in Radio City through these cast iron sleeves which are a part of the seating equipment. A small grille fitted into the upright of each seat is placed over the sleeve, and air is drawn through them into the plenum chamber beneath the floor. Sleeves are cast in two sections. In the illustration the lower sections have been set in the forms for the concrete sub-floor preparatory to pouring the masonry. The upper sections will be placed over these and levelled up before the finishing floor surface is laid. A small damper in the individual sleeves regulates the rate of exhaust.



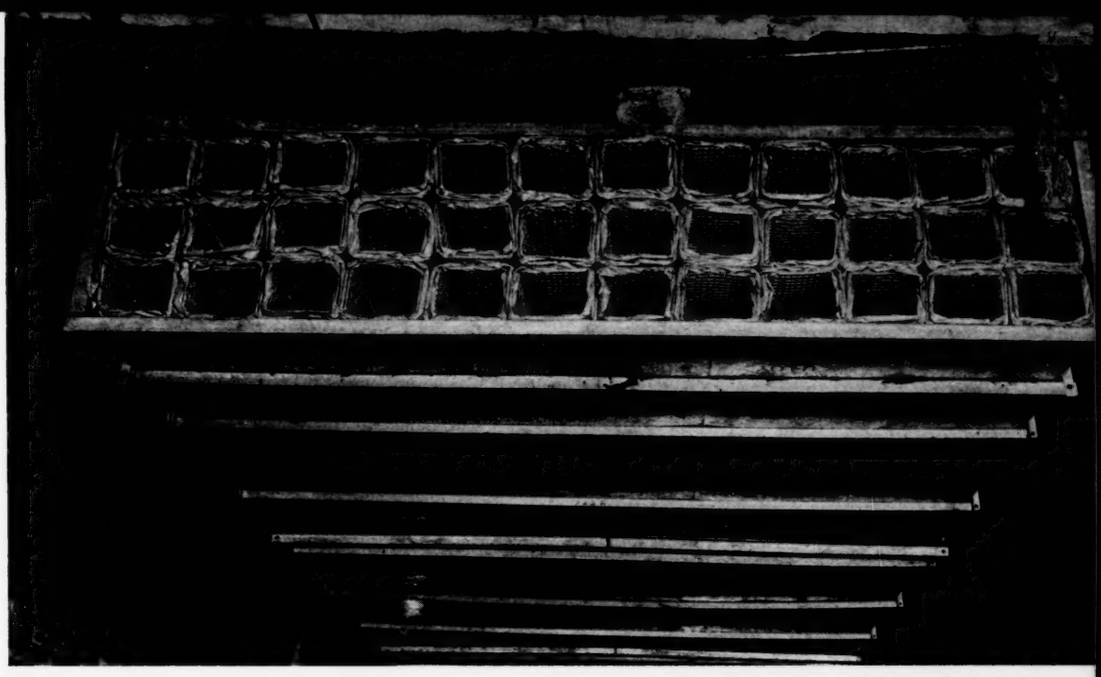
Relief air louvers through which a quantity of air passes directly to out of doors from the main orchestra by virtue of the pressure maintained in the theater above outdoor barometer. This relief is soundproofed from exterior noise disturbances.



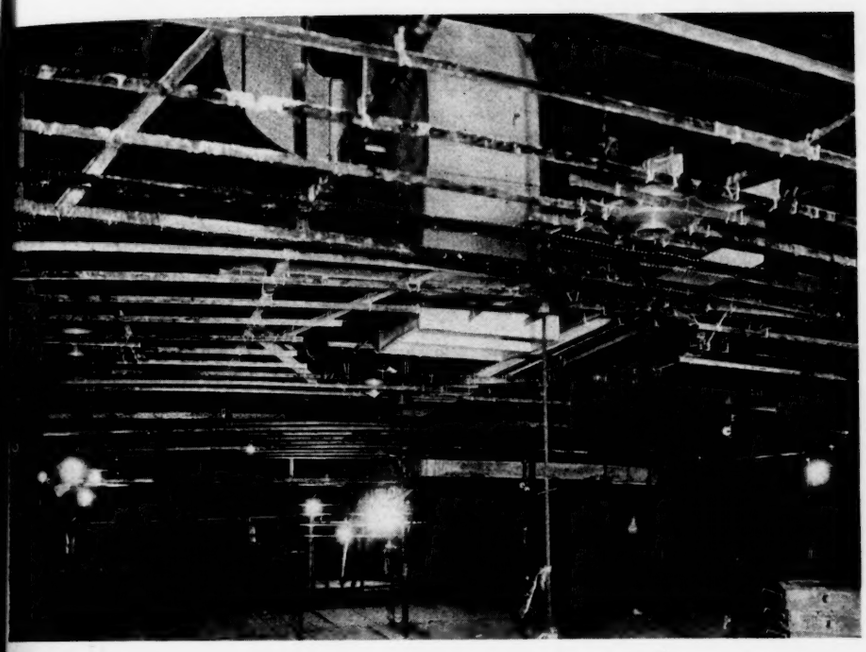
Installing the spray heads in one of the dehumidifiers in the sound picture theater, which will seat 3500 persons. The refrigerating equipment in the theater will have a capacity of 225 tons.

Center—V

apid rate in Rockefeller Center, uilding project. When com- ontain the world's largest office he world's largest theater, and he heating and air conditioning ound Picture Theater and the tudios, part of this project.

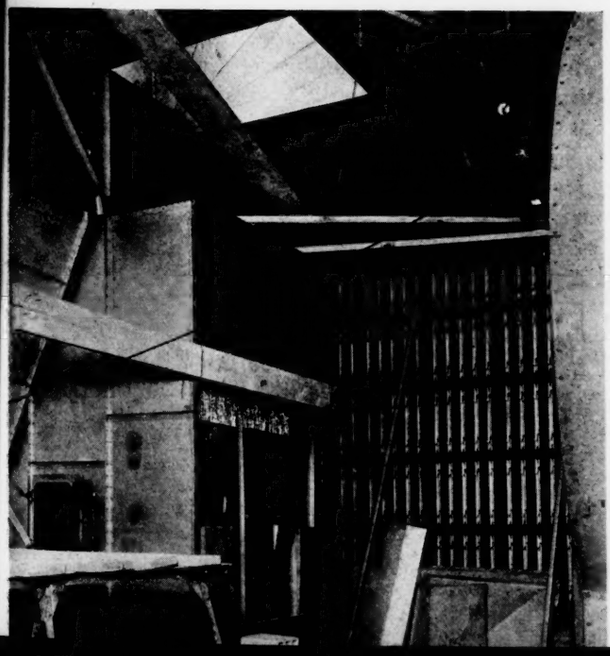


Sound absorber in duct connection to orchestra plenum in the sound picture theater showing paper covered insulated tubular cells for sound absorption. The insulation is made of dried sea weed.

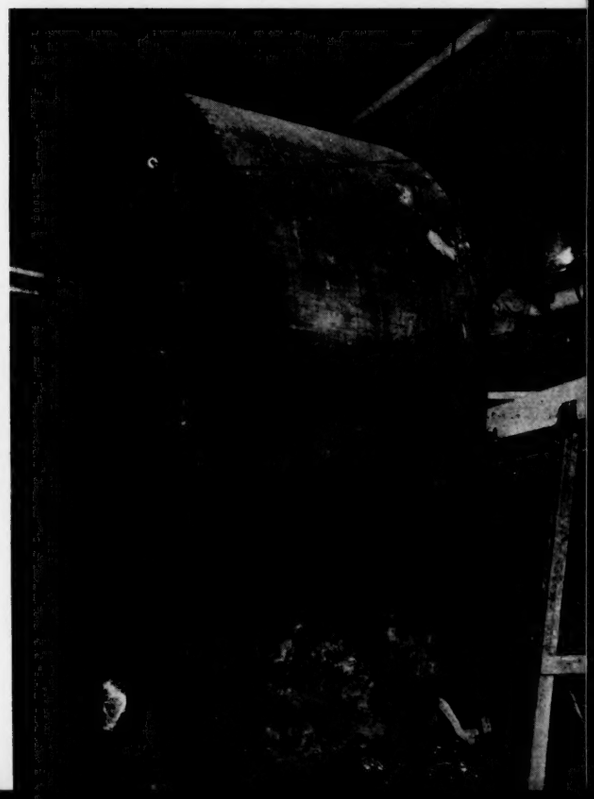


Frame work which will support the world's largest chandelier, 25 ft. in diameter, in the R.K.O. Sound Picture Theater. The 400 flood lights and their great lenses emit an amount of heat equivalent to 1470 sq. ft. of radiation. For that reason a special ventilating system, a part of which can be seen in the illustration, will carry off the heat. Supplementing the indirect light that is to come from the large central fixture, there will be a unique system of direct lighting which is accomplished by powerful beams of light that are shot down through 200 minute holes in the ceiling. It will thus be possible, by crossing the upward beams of light from the central fixture with the downward beams of light from the small ceiling holes, to produce an unlimited number of color blending effects.

Details at dehumidifier showing filter prior to completion of sheet metal casing connections in the Sound Picture Theater.

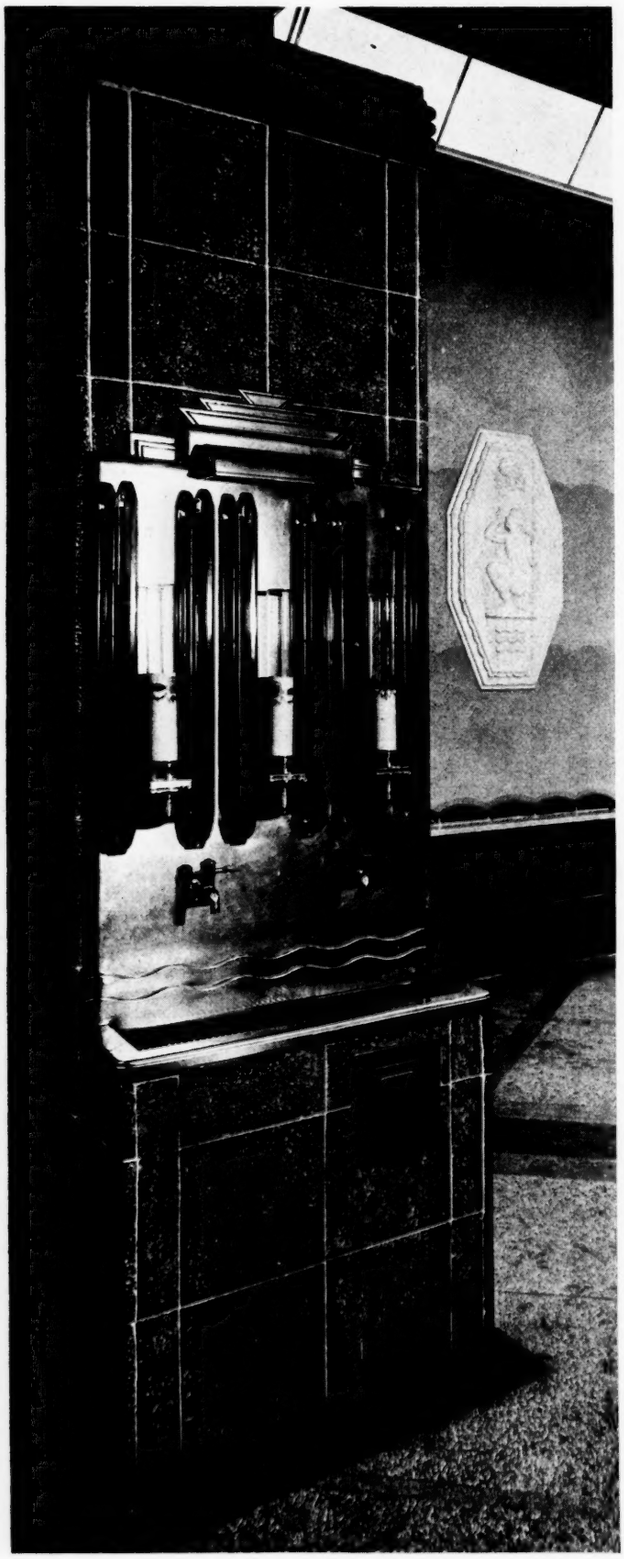


One of the giant fans which will supply five tons of conditioned air per minute to the sound theater. In the background steamfitters are shown welding pipe which will carry water to the dehumidifiers.



Piping and Refrigeration for W

THE first consideration in planning for a supply of cool drinking water is an adequate and suitable source which must not be contaminated after entering the system. This is no different from the requirements for ordinary drinking water supply, except that where the added refinement of cooling is used it involves enough additional cost so that pipe sediment or other undesirable characteristics in the water should be even more carefully guarded against. It is also very desirable that the cooled water supply shall be really cool, and that it shall be available without running large quantities of water to waste. Moreover, enough care should be given to the design so that neither a veritable shower of water nor a mere trickle will discharge from fixtures when they are opened to draw water.



Cooling of the water is commonly accomplished by the use of mechanical refrigeration, and the compression cycle is the one commonly used.

As in so many cases in the use of refrigeration, the cooling of water may be accomplished either through the use of individually-operated units or through central systems. The unit plan is comparatively recent, but is becoming increasingly popular. The central system, while older, is considered especially suitable in the case of large structures or groups of buildings. The individual unit installation with a small refrigerating plant built into it is popular for cooling water for individual offices, and where the number of people being served is small.

The unit systems are commonly built into compact cabinets and are not further considered here.

Central systems, such as are so often used in large buildings, can be piped in a number of interesting ways.

Piping Systems

Fig. 1 shows diagrammatically a simple arrangement for a small system. This consists of an open type cooling tank or interchanger located in an elevated position above the fountains, to which the water runs by gravity. This is known as an open system.

Where the city water pressure is high enough to deliver water to the fountains, the arrangement shown in Fig. 2 may prove feasible, but a closed water cooler must be provided instead of the open one used in Fig. 1. Neither of these methods would be expected to prove very satisfactory in operation. Their chief asset is a cheap installation cost. Use of these methods usually involves a noticeable waste of water, as it is often necessary to draw several quarts of water before it becomes cold at the fountain. The cold water standing in the lines after leaving the cooler tends to warm up each time between uses.

Fig. 3 shows a closed system using a circulating pump and a pressure type water cooler, to which city water is connected directly. During periods when the tap water is sufficiently cold, the refrigerated cooler and circulating pump may be by-passed and the fountains supplied direct with city water.

Where the minimum city pressure is sufficient to supply the entire building, the pump merely has to overcome the velocity head, which is negligible.

Water-Cooling Systems

By R. B. REAGAN

For buildings of moderate height, and where the system is expansive, the piping method shown in Fig. 4 is popular. It includes both a circulating pump and an open type water cooler.

Fig. 5 shows a method used for higher buildings than indicated by Fig. 4. It is provided with a pressure type water cooler and circulating pump.

In case there is an existing house tank at a higher level than the circulating system, then the closed purge tank can be replaced by a connection from that point to the house tank, which would provide both the make-up and the necessary air vent.

For higher buildings, a double system with each part similar to that shown in Fig. 4 is often used. As may be noted from the sketch, Fig. 6, the water coolers and pumps for both of the systems are placed together in the basement. The requirements and design of each system are identical for systems in buildings of half the height, excepting for the increased pressure under which the cooler, pump and lower half of the vertical piping of the upper system will operate.

As the lower part of the upper system has no service connections, the water in both supply and return risers balances. Therefore, the load on the pumps required for circulation of the water does not necessarily increase with the height of the building, but is determined rather by the amount of friction which must be overcome in circulating the water.

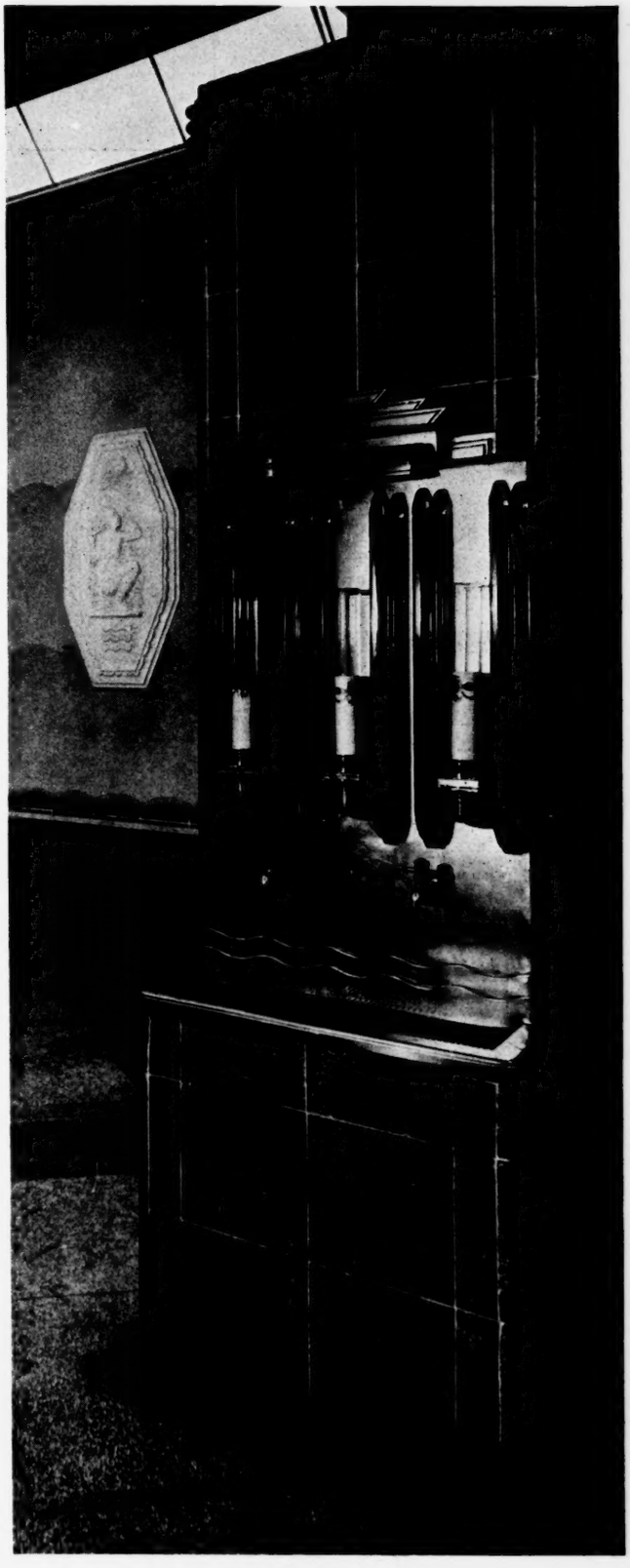
The piping system should be so designed that every fountain when opened may deliver at once water at a uniform, specific temperature. Where conditions permit it is best to serve each group of fountains on separate circuits, each of which is connected to feed and return lines.

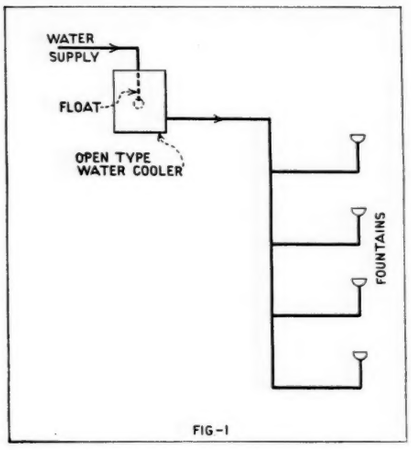
The number of shut-off valves may be limited for the sake of economy of installation, as their insulation involves a considerable cost. The advantage in using plenty of such valves lies in the fact that they permit repairs to be made with a minimum of interference. In a well-constructed installation repairs should be necessary only at infrequent intervals.

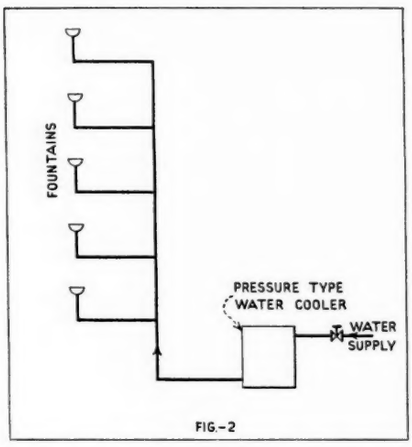
Also, in a multi-story building it is desirable that the drinking fountains on the various floors be located vertically one above the other. Long dead ends and connections so made that they will trap air should be avoided.

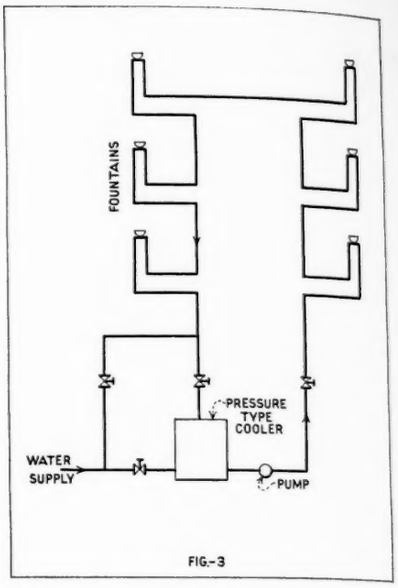
The nuisance of having to run warm water from an uncirculated connection, or getting an air blast through the fountain preceding or during the taking of a drink, is an all too common annoyance. A little care and only slight additional expense will eliminate it. A few feet of return piping is usually all that is necessary, as this makes a completely closed system.

Points of importance in selecting a desirable drinking fountain are: its fitness for the service required, whether it is for pedestal or wall mounting, single or multiple outlet, and if the emission will be of liberal volume. Also, the nozzle should be so arranged that the lips of









Figs. 1, 2 and 3. A simple open system, a simple closed system, and a closed system with circulating pump

the drinker cannot come in contact with it, and the faucet should be substantial and self-closing.

The cooling apparatus and circulating piping should be properly covered with insulation.

Where a coil-tank type water cooler is selected in place of a water interchanger, a means must be provided to promote agitation of the water in the tank, otherwise the water is not evenly cooled, and failure of the automatic control devices would cause a freeze-up.

Refrigeration Load Required

As an example of a calculation, assume that we are to provide a drinking water system for a typical manufacturing plant with the following characteristics:

(1) 350 workers, exposed to heat incidental to metal working and using a water consumption of 1 qt. per man-hour.

(2) Total distribution system to the various drinking fountains will use 2500 ft. of pipe line.

(3) Average shop temperature during the peak season is approximately 90°.

(4) The available water supply from city mains averages 80° and at 50 lb. pressure.

(5) It is desired to supply drinking water at any of the fountains at a temperature not exceeding 50°, and a temperature rise not to exceed 5° throughout the circulating system, which is insulated.

From the data we determine that the circulating water will have an average temperature of 47.5°; that

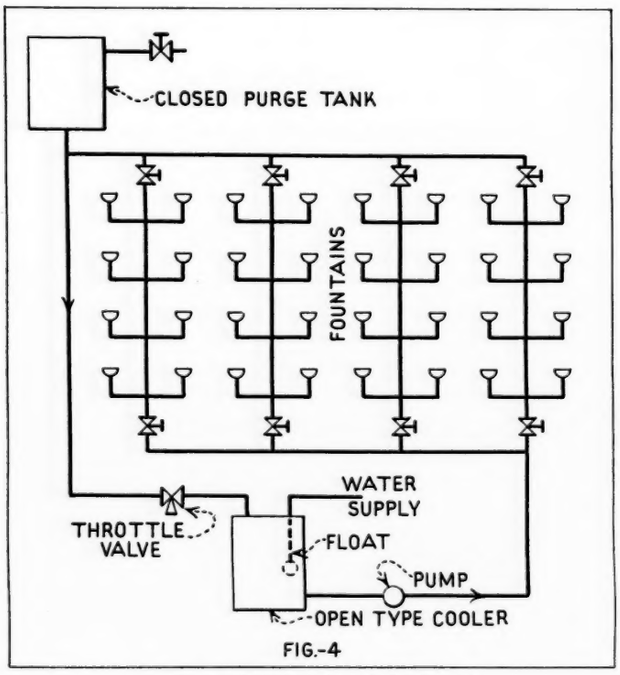


Fig. 4. System for building of moderate height

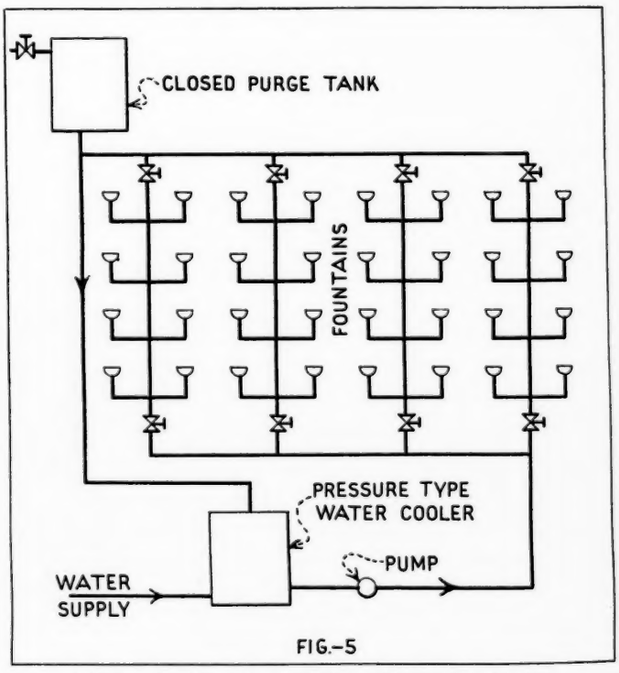


Fig. 5. System for building of greater height than that of Fig. 4

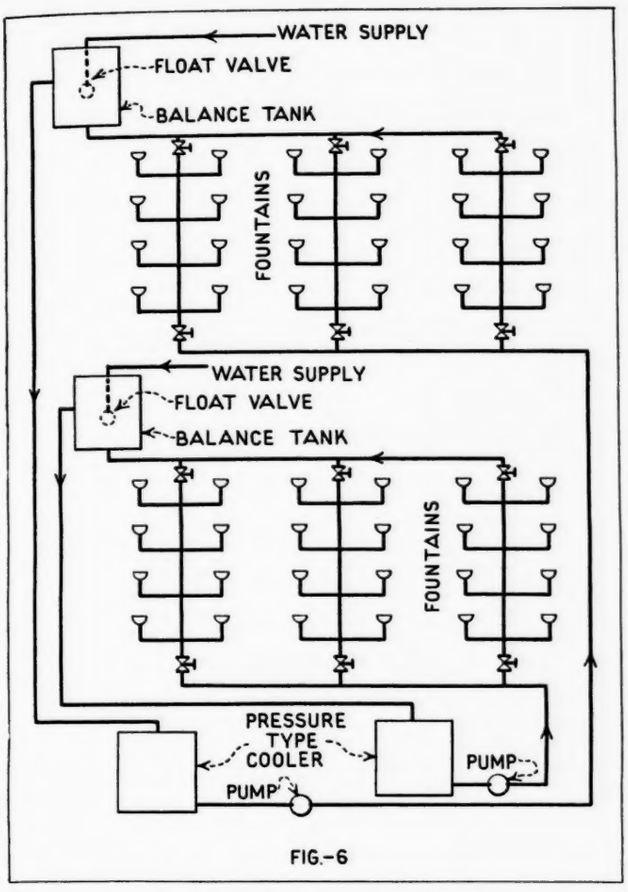


Fig. 6. System for tall buildings

the difference between room and water temperature will be 90° — 47.5°, or 42.5°; and that the heat loss from the insulated piping will be 0.152 B.t.u. per lin. ft. per hr. per degree difference of temperature.

With these factors available, solve for the quantity of water circulating:

$$\text{(a)} = \frac{2500 \times 42.5° \times 0.152}{5 \times 62.5} = 51.68 \text{ cu. ft. per hr.}$$

$$\text{(b)} = \frac{350}{4 \times 7.48} = 11.7 \text{ cu. ft. per hr.}$$

$$\text{Total} \ldots\ldots = 63.38 \text{ cu. ft. per hr.}$$

The water estimated to be consumed at the fountains, therefore, will equal:

$$\frac{63.38}{60} = 1.06 \text{ c.f.m.}$$

$$\text{or } 1.06 \times 7.48 = 7.93 \text{ gal. per min.}$$

In the preceding calculation the quantities imposed have the following values:

2500 = total length of pipe, feet
42.5° = temperature differential between air and water
0.152° = heat loss through insulation, B.t.u. per lineal foot per hour per degree
5° = possible water temperature rise
62.5 = pounds of water in one cubic foot
60 = minutes in 1 hr.
7.48 = gallons of water per cubic foot

To maintain the predetermined temperature in the circulating water line, calculate for the necessary refrigeration tonnage:

$$\text{(a)} \quad \frac{2500 \times 42.5° \times 0.152}{12{,}000} = 1.34 \text{ tons}$$

To cool make-up water:

$$\text{(b)} \quad \frac{11.7 \times 62.5 \times (80° - 45°)}{12{,}000} = \frac{2.14 \text{ tons}}{3.48 \text{ tons}}$$

To offset mechanical heat of the pump:
(c) Add 6% to calculated tonnage = 0.20 tons

To care for loss through cooler:

$$\text{(d)} \quad \frac{1.34 \times 45 \times 4}{288{,}000} = 0.08 \text{ tons}$$

To care for contingencies:
(e) Add 8% to 10% = 0.34 tons

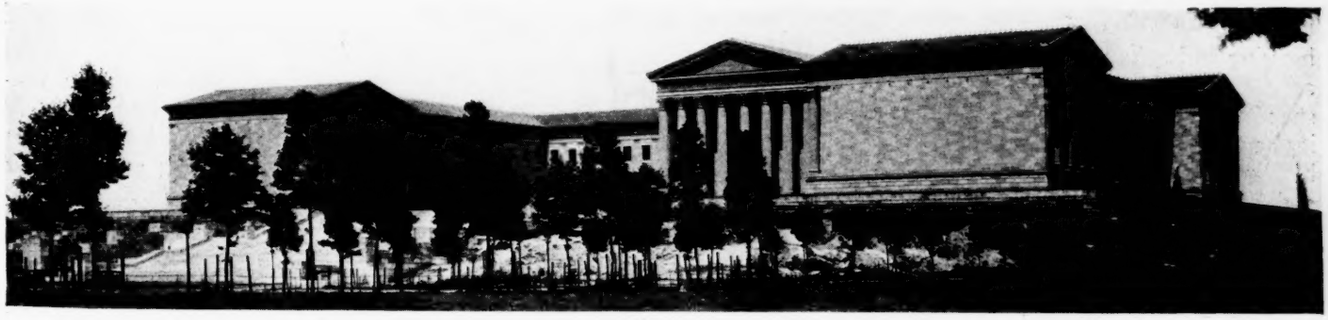
Total refrigeration required, 4.10 tons.

The quantities above have the following values:

12,000 = one ton refrigeration effect, per hour, in B.t.u.
11.7 = cubic feet of water required. (b)
80° — 45° = difference between city and fountain water temperature
1.34 = tons of refrigeration to care for pipe losses. (a)
45° = circulating water temperature
4 = quarts per gallon
288,000 = one ton of refrigeration per 24 hr. in B.t.u.

The size of the recirculating pump to be selected is readily obtained when the quantity in gallons per minute has been determined, plus the total head in feet and the friction losses that the entire piping system offers.

There frequently arises a tendency to minimize sizing of the distribution piping, which results in higher velocity of flow and necessitates an increase in power output. On the other hand, the alternative of lowering the circulating pressure by installing oversize piping has the decided disadvantage of increasing the refrigeration demand. This problem may be best answered by installing recommended sized piping to care for the imposed load requirements, even though its use results in a trifle higher initial cost.



Galloway

Philadelphia Museum of Art. Conditioned air is supplied to those portions of the building where old paintings and fabrics require carefully controlled atmospheric surroundings

Ventilation of the East Boston Traffic Tunnel

By A. A. BERESTNEFF

The East Boston Vehicular Traffic Tunnel, now under construction, connects Boston, Mass., with East Boston, and passes beneath the main ship channel of Boston Harbor. It is the fourth vehicular tunnel to be built in this country. The total length of the tunnel is 6150 ft.; the distance between portals being 5650 ft.; the maximum depth of the roof below mean low water is 54 ft.; the outside diameter of the circular cross-section is 31 ft.

INVESTIGATIONAL work done in connection with the design of the Holland Tunnel, the first vehicular tunnel, showed that an upfeed ventilating system gave better results than the downfeed arrangement. The reason was that, with the latter scheme, the heated air from the exhausts of the motor cars tended to rise, opposing the air stream from the ventilating system. The noxious gases would rise above the breathing level and then be forced down by the air stream, with the result that the poisonous gases passed the breathing zone twice, while the upfeed system allowed such gases to pass the breathing line only once. The same investigations showed that the maximum allowable amount of CO is four parts per 10,000 parts of air, by volume.

These principles and others set up from the studies made in connection with the Holland Tunnel studies were the basis for the design of the ventilating system for the East Boston Tunnel.

The length of the Boston tunnel is 5,400 ft. between the portals. Between the ends there are two ventilation buildings in which the blowing and exhaust fans are located. The fresh air is forced into the tunnel in a longitudinal direction through a "fresh air" or "blowing" duct located at the bottom of the tunnel section underneath the roadway.

Vitiated air is removed through the exhaust duct located at the top of the tunnel section above the ceiling. Along their whole lengths both ducts are divided into four independent parts—one from the entrance to a point under the ventilation building, another from under the building to the middle of the river, and the other two similarly on the other end of the tunnel. The areas of the blowing and exhaust ducts are 91 and 100 sq. ft. respectively. The air velocity at the beginning of the blowing duct and at the end of the exhaust ducts is about 11,500 f.p.m. at the time of maximum air supply.

Fresh air is forced by blowers into the blowing duct. The air is discharged into the tunnel through ports provided every 15 ft. on both sides of the duct. As shown in the detail, the air from the flue enters a so-called air expansion chamber. The air from the flues expands in the chamber and escapes into the tunnel through a slot, the width of which can be regulated by a special face plate. This controls the amount of fresh air supplied into the tunnel. Another regulating damper is located at the entrance to the flue. The purpose of this damper is to allow the air to be discharged in equal quantities through all flues. Once adjusted the position of slide bumpers will be kept constant for any amount of air supplied to the tunnel.

Exhaust ports are located in the ceiling. The reason for the 45° angle, shown in the section of these ports, is that at this angle the losses in the exhaust ducts were found to be at a minimum. The exhaust ports are also provided with adjustable slides for regulating the amount of air exhausted through each port.

This arrangement of ports and expansion chamber results in a relatively small outlet velocity through the continuous slot. The air motion is gentle, and even when the air quantity is such as to change the air in the tunnel 40 times per hour, the draft is hardly realized by a person standing on the sidewalk just above the discharge opening

Ventilation Buildings and Fans

There are two ventilation buildings provided for housing the fans for each part of the tunnel. The location of the blowing fans in the East Boston Ventilation Building is shown on Fig. 2. There are seven fans all working on the same plenum chamber. The air intakes are located at the second, third and fourth floors of the ventilation building. As seen in the drawing, the fans are located on the first three floors of the ventilation building, which consists of four stories each about 18 ft. high.

A special partition divides the lower part of the chamber into two parts,

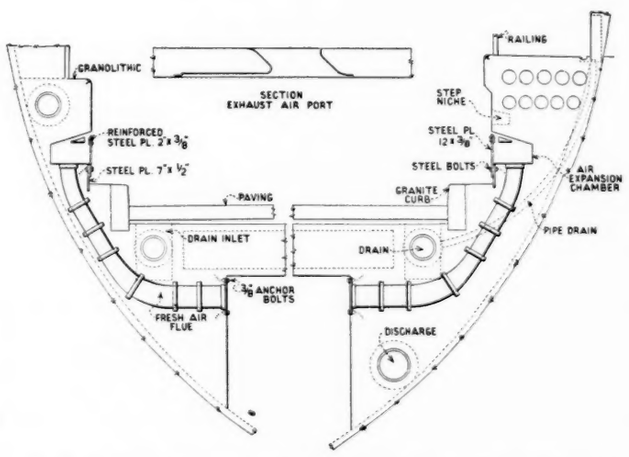


Fig. 1. Section through tunnel showing arrangement of air supply

each leading to the corresponding blowing duct for distribution of fresh air along the tunnel through the outlet ports as described above. At the top of the partition there is a special damper provided for regulating the amount of the fresh air forced into the blowing ducts on each side of the building.

The outlets of all blowers are connected to the plenum chamber through the short ducts ending by the openings, the area of which can be regulated by special rotating dampers. If necessary the area of the openings can be made larger or smaller. The openings can be also shut off entirely. These two adjustments regulate the amount of air supplied by the fans.

Fig. 3 shows the exhaust fans for the same ventilation building. These fans are located on all four floors of the building. The two exhaust ducts entering into the common chamber are separated by a partition at the top of which there is a rotating damper, by means of which the exhaust air flows are regulated. Each of the exhaust fans is located in a small chamber separate from all other fans. This chamber is connected with the common plenum chamber by an opening which can be varied or shut off entirely by means of a special rotating damper for controlling the amount of air exhausted.

Table 1 gives some data concerning the supply and exhaust fans as listed in the specifications. As shown in the table, the total amount of air supplied by all supply fans is 1,095,000 c.f.m. The total amount of air handled by exhaust fans is slightly larger, being 1,119,000 c.f.m. The reason for the difference is due to the difference in densities and amount of the air handled by the blower and exhaust fans.

TABLE 1

	Total Number of Fans	Number of Fans Full Load	Quantity of Each Fan, C.F.M.	Total Quantity, C.F.M.	Total Pressure at fan In. of Water
Boston Shaft					
Blowers	7	6	93,000	558,000	2.20
Exhausters ..	7	6	95,000	570,000	2.50
East Boston Shaft					
Blowers	7	6	89,000	537,000	2.25
Exhausters ..	7	6	91,000	549,000	2.70

Note: The pressures have been figured on a basis of air densities of 0.08 lb. per cu. ft. for blower fans and 0.0784 lb. per cu. ft. for exhaust fans.

The air quantities, however, as well as total pressures given in the tabulation have been based upon the operation of six fans on each duct, the seventh fan being reserved as a spare.

Since seven fans will be connected to one duct in

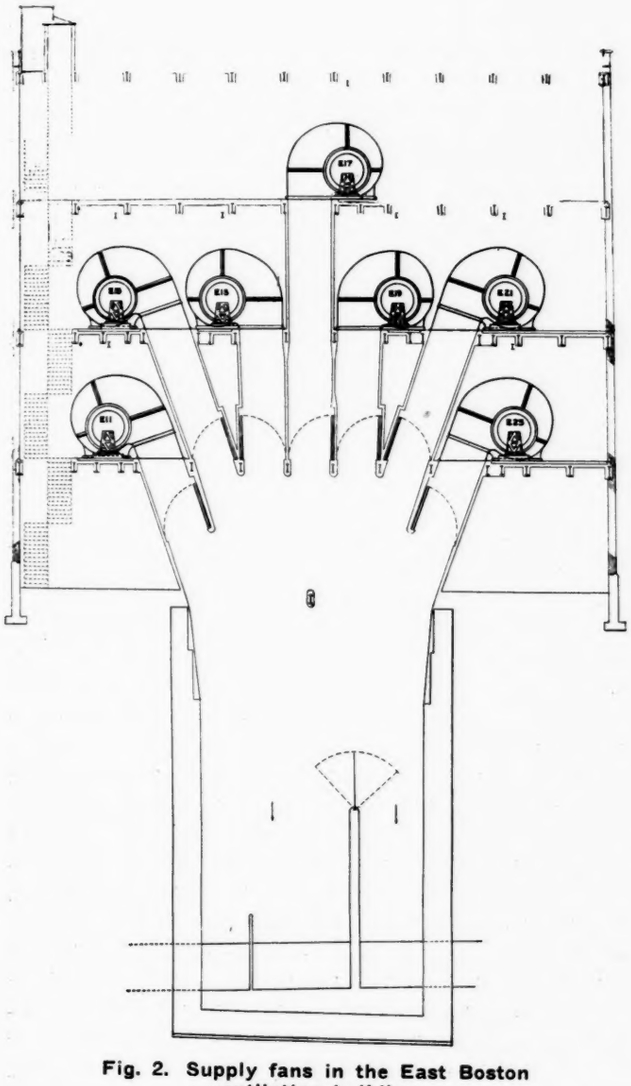


Fig. 2. Supply fans in the East Boston ventilating building

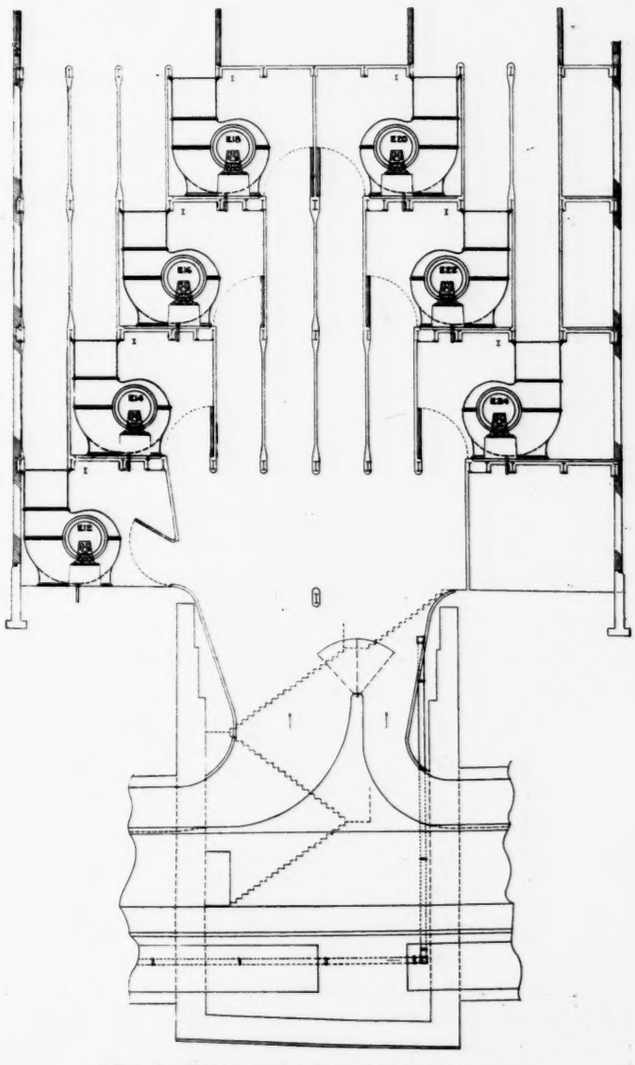


Fig. 3. Exhaust fans in the East Boston ventilating building

each of the ventilation buildings, it is necessary that these fans should have characteristics which will permit them to operate successfully in parallel. The horsepower characteristics must be such as to prevent overloading of the motors under operating conditions. It is also essential that the fans maintain a satisfactory efficiency over a wide range of volume. The specifications call for a fan whose static and total pressure curves shall descend gradually from approximately one third wide open volume to full wide open volume.

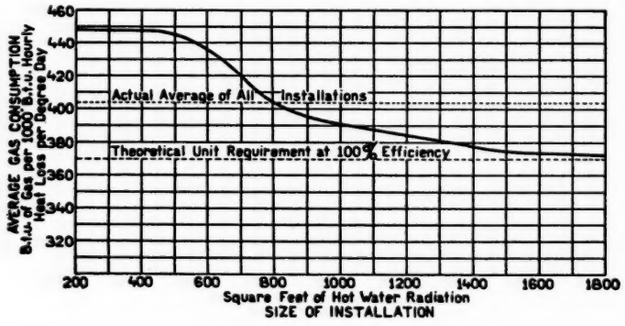
The writer wishes to express his indebtedness to Col. Thomas F. Sullivan, chairman, and E. R. Springer, chief engineer, Boston Transit Department, for very valuable cooperation in the preparation of this article.

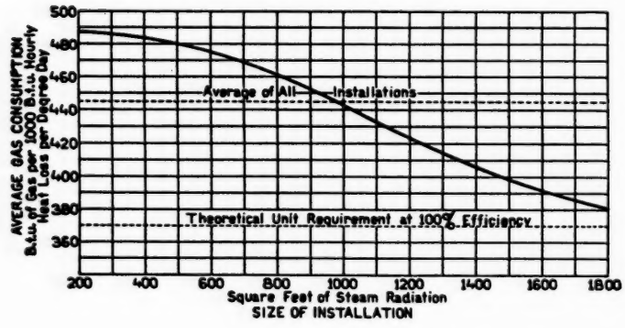
Unit Gas Consumption Data

THE actual results obtained from the present survey of 700 private residences averaged as follows:

Type of Installation	B.t.u. per degree day per 1000 B.t.u. hourly heat loss from the house
Gas-designed hot water boilers	404
Gas-designed steam boilers	445
Converted hot water boilers	452
Converted steam boilers	503

These figures show overall seasonal heating efficiencies from a maximum of 91.6% to a minimum of 73.5%. They also show that the overall results of hot water equipment are 10% better than steam equipment and that the difference between the average efficiency of all gas-designed versus converted equipment was about 12% in favor of the gas-designed equipment.





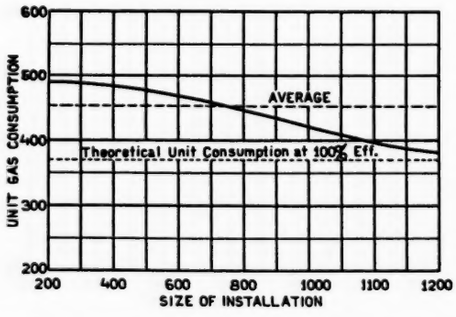


Fig. 1 (top). Unit gas requirements for heating residences with gas-designed hot water boilers. Fig. 2 (center). Unit gas requirements for heating residences with gas-designed steam boilers. Fig. 3 (below). Unit gas requirements for heating residences with automatic gas burners in water boilers.

Fig. 1 shows the relationship between the size of the installation and the average gas consumption, expressed in B.t.u. of gas required for each 1,000 B.t.u. hourly heat loss from the building, figured in accordance with the methods given in the *Guide,* per degree day. This figure refers to the results obtained for gas-designed hot water boilers. These data are all for private residences. As the size of the job increases the unit gas requirement seems to decrease.

Fig. 2 is a similar plot based on gas-designed steam boilers. Fig. 3 is another similar plot based on converted hot water boilers.

When the lists of jobs were studied in detail in any one size class, it was found that individual installations showed extremely wide differences in efficiency. It showed that there was a small group of gas-designed jobs that was better than the best of the conversions and it also showed that a small group of converted installations was less efficient than the least efficient of the gas-designed equipment. The majority of the jobs was on a par as far as efficiency is concerned; the 12% difference in average efficiency was largely caused by the installations which fell into the highest and lowest groups.

Eliminating freaks, the worst of the converted installations averaged about 558 B.t.u. per degree day per 1000 B.t.u. hourly loss from the building, as contrasted with 525 for the worst of the gas-designed.

The extraordinary part of these data was the unit figures obtained from the best of the installations. The best of the gas-designed jobs show about 250 B.t.u. per degree day per 1000 B.t.u. hourly heat loss from the building, and the best of the converted installations about 274 B.t.u. Taking the two unit figures just mentioned as an example, they indicate that the overall seasonal thermal efficiencies were 140% and 150% respectively. In other words, the theoretical heat losses, calculated according to recognized principles, and making due allowance for every factor which was known to be involved, were found to be one and one-half times as great as the heat input furnished by the gas—*C. George Segeler in "Automatic Gas Burners," a paper presented before the A.S.H.V.E. summer meeting, June,* 1932.

Modern Ventilation for Stowage Space in Freighters

ON the six fast vessels recently introduced in the Antwerp-Rotterdam trades by the American Diamond Lines, there has been installed a system of mechanical ventilation which provides probably the most efficient method yet devised for the positive and thorough ventilation of cargo compartments on freight ships.

This advanced type of cargo ventilation has been designed and adopted after a thorough study of all the factors entering into ventilation, embracing cargo compartment temperatures under operating conditions at sea with reference to atmospheric temperatures, humidity, sea temperature, etc., and is in recognition of the inherent deficiencies found in the usual method of ventilating ships' holds.

Given proper stowage, fresh fruits, flower bulbs, provisions, etc., may safely move in ordinary stowage. If and when damage occurs, faulty ventilation is usually found responsible. One form of damage is that of sweating, the accumulation of moisture and its subsequent condensation on the sides and deckhead of the cargo compartment. Another form, and perhaps the more serious, is that produced by cargo gases and fumes which, if allowed to accumulate in the compartment due to imperfect circulation of fresh air, affect the cargo itself. In the case of apples, by way of illustration, the fruit is subject to damage by scald—a discoloration produced by gases generated by the fruit itself.

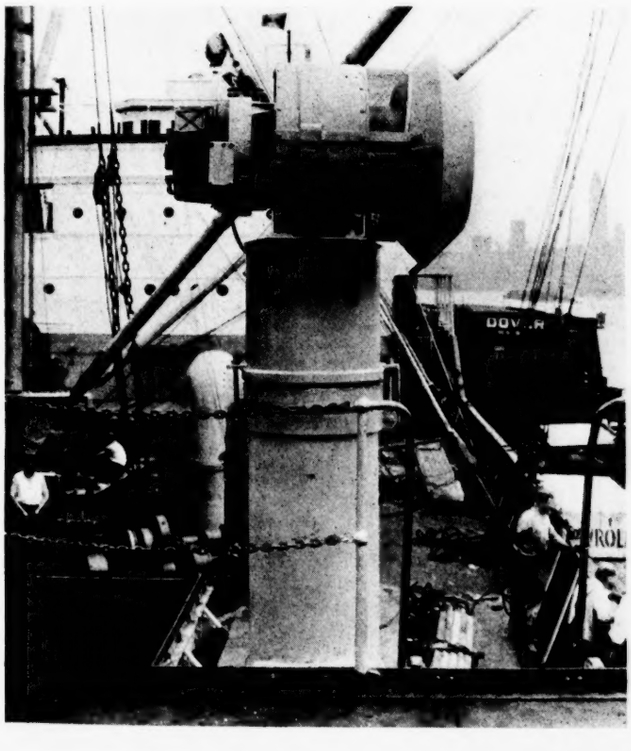
The ordinary method of ventilating ship's cargo spaces is by cowl ventilators, serving all the holds and 'tweendeck spaces. The principal defect of such a system is that it may be uncertain in its operation. It is the simplest form of ventilation and does not provide for positive, complete and constant ventilation. The penetration of the cargo compartment with air is dependent altogether on natural air forces which are variable in their intensity. Thorough ventilation is only possible when the air is forced into the compartment under positive pressure. Satisfactory ventilation is ventilation which produces a rapid and vigorous and well distributed circulation of the air in the compartment so that the flow of the fresh air causes complete air change to take place rapidly and continuously.

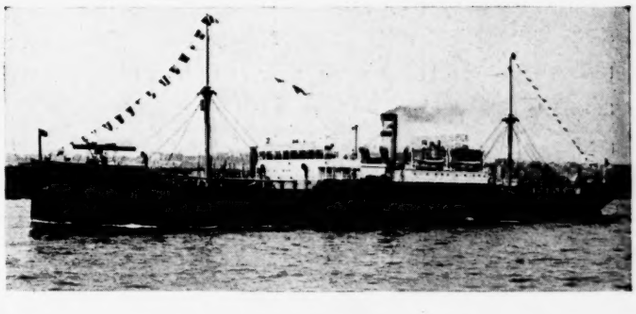
The American Diamond Line's system gives this thorough ventilation. Each cargo compartment is equipped with two motor-driven ventilating units, located on the ventilating trunks. A single unit has capacity to change the air in a cargo compartment every 15 min.

Outlet ventilating trunks located on the bulkheads at a considerable distance from the mechanical inlet ventilators, insure the subjection of the entire compartment and its cargo to the action of the incoming air streams.

Positive distribution of the air across the air inlet end of the compartment is effected by use of large canvas distribution pipes having numerous outlet nozzles. Location of these pipes and hence the direction of ventilation can be arranged to suit cargo conditions and stowage.

Sweating is eliminated by this mechanical ventilating system, since the rapid, positive circulation of air throughout the cargo and cargo spaces maintains a much more equable temperature balance between the outside atmosphere and that of the cargo compartment. Sea spray cannot enter the cargo compartment, as so frequently happens in the use of ordinary cowl ventilators. The design and construction of the motor-driven ventilator units is such that they can be trimmed or rotated to any position to suit wind and weather conditions encountered at sea. Their continued operation is thus assured.—*M. J. Hanlon, Operating Manager, American Diamond Line, in "Shipping Digest."*





(Below) The S. S. Black Hawk, of the American Diamond Lines, one of the six vessels recently reconstructed for these lines and which will deliver cargo from New York to Antwerp and Rotterdam in 10 days. All the six vessels have mechanically ventilated cargo compartments. (Left) Mechanical ventilating unit on deck of S. S. Black Hawk.

Construction of a Simple Psychrometric Chart

II—Conclusion

By A. E. BEALS

THE completed chart, with effective temperatures as dotted lines, is shown by Fig. 2, reproduced on the accompanying folded insert. Included also with Fig. 2 are two supplementary curves. These curves show the volume occupied by 1 lb. of pure air without moisture and the volume occupied by 1 lb. of pure air when the space so occupied is also filled with vapor at the saturation point, each curve being plotted for various temperatures.

It will be noted that the curve for dry air is a straight line, the volume varying directly as the absolute temperatures. Likewise the volume of 1 lb. of saturated air will increase directly as the absolute temperature due to accession of heat, provided there is no addition made to the moisture content.

Therefore, a straight line, drawn parallel to the dry volume line from any point on the saturated volume line, will represent the change in volume from the saturated condition when warmed up to any higher temperature.

These supplemental volume curves are illustrated in Fig. 1. Let it be assumed that it is required to determine the volume per pound of dry air for a mixture of air and vapor at 80° dry bulb and 60° wet bulb temperatures.

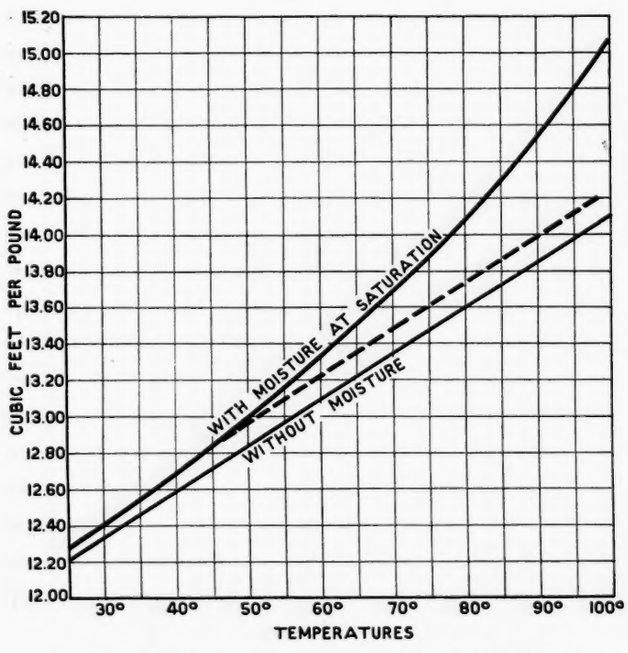


Fig. 1. Supplemental curves showing variation in air volume with change in temperature

Referring to the heat-moisture chart, Fig. 2, and projecting horizontally to the saturation curve the intersection of the 80° dry bulb and the 60° wet bulb temperature lines, it will be found that the dew point for this condition is 46°.

If a line, starting from the 46° point on the saturated volume curve, is drawn parallel to the dry volume line, its intersection with the line of 80° temperature will indicate the required volume for the observed temperatures. In the assumed case this volume is 13.73 cu. ft. per lb. of dry air mixed with 45.92 grains of water vapor.

All problems relating to air conditioning are concerned primarily with changing either the heat or the moisture content and usually both. An important feature of the chart is that the extent of these changes, proportioned to 1 lb. of dry air under given conditions as to temperature and moisture content, can be indicated upon this chart by means of straight lines drawn between any two points.

In order to plan and design an air conditioning unit which shall successfully maintain a desired condition as to temperature and moisture content two essential data facts must be determined. These are the total amount of heat from all sources which flows into the space in question and the amount of water vapor injected therein. It is important that in the total amount of heat to be reckoned with the latent heat of evaporation of the moisture at the desired temperature of the space must be included.

Example 1

Let it be assumed that it is desired to maintain within a certain space a dry bulb temperature of 70° and a relative humidity of 50%. The amount of heat to be removed amounts to 1,440,000 B.t.u. per hr., while 2,250,000 grains of moisture must be removed per hour.

* * *

From the steam tables it is found that the vapor tension at 70° temperature is 0.3626 lb. per sq. in. Therefore, the vapor tension at 50% relative humidity will be 0.1813 lb. per sq. in. This is found, also from the steam tables, to be the pressure of saturation at 50.48° temperature. The heat-moisture chart shows that 50.48° is the saturation temperature, or dew point, for all temperatures of a mixture of 1 lb. of dry air and 54.4 grains of water vapor (see Fig. 3).

Projecting the point of 50.48° temperature, lying on the saturation curve, horizontally to the point A on the 70° dry bulb temperature line, it will indicate that

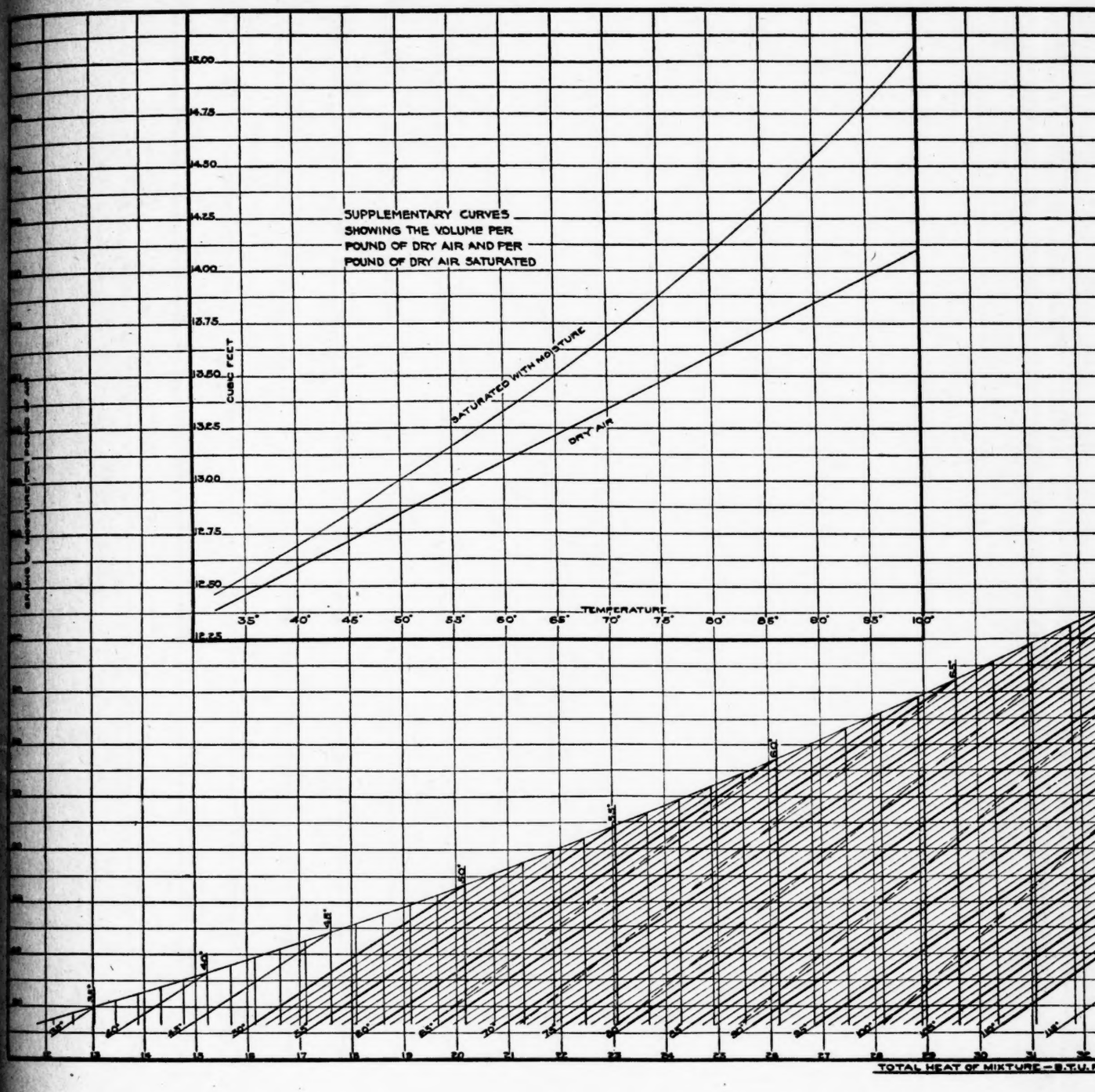


FIG. 2. THE BEALS

The barometric pressure for the saturation curve is taken at 29.92 in. of mercury or 14.7 lb. per sq. in. of absolute pressure.

The moisture content and the heat content scales are each in terms of 1 lb. of pure air contained in the mixture.

To find the dew point of any condition: Project the point of the given condition horizontally to the saturation curve and note the temperature thus indicated. For example, assume 80° dry bulb and 60° wet bulb temperatures. Locate the intersection of the oblique line representing 80° dry bulb temperature with the vertical line representing 60° wet bulb temperature and follow horizontally from this intersection to the saturation curve and find 46° as the required dew point.

To find the maximum amount of moisture possible for a given temperature: Follow the obli
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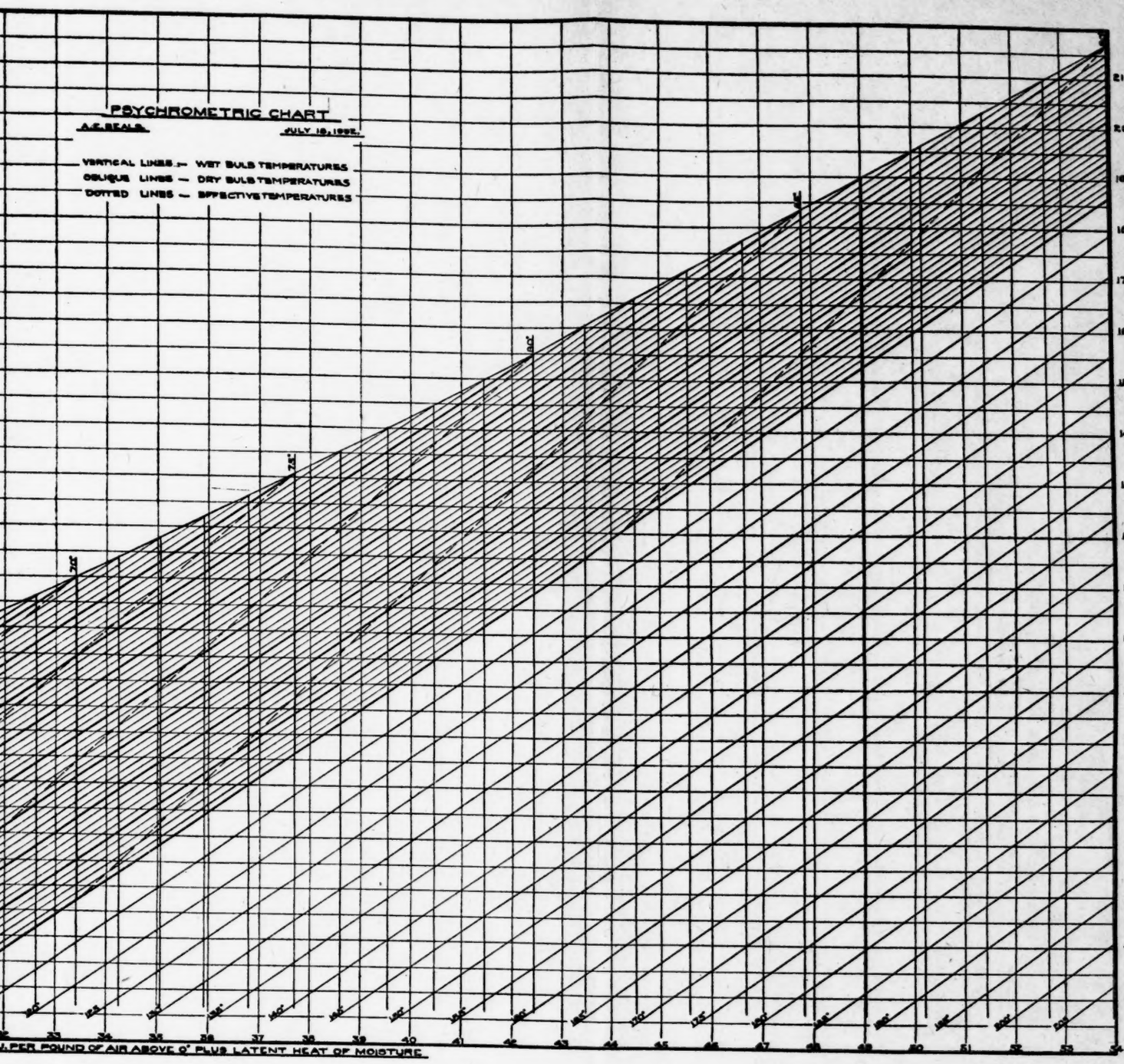

PSYCHROMETRIC CHART

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To determine *approximately* the per cent of relative humidity for any given condition: Divide the grains of moisture per pound of air for the given condition by the maximum grains possible for the dry bulb temperature of the given condition. The value thus found will be from 97% to 99% of the true relative humidity for temperatures between 60° and 90°.

The resultant of the mixture of two bodies of air will lie upon a straight line joining the coordinate points representing the conditions of the bodies. The coordinate point of the mixture will divide this line into parts, the lengths of which will be inversely proportional to the weights of the two bodies of air.

58.3° is the wet bulb temperature required to be maintained, and that the heat content will be 25.06 B.t.u. per lb. of dry air.

The ratio of the 2,250,000 grains divided by 1,440,000 B.t.u., or 1.56, will be the number of grains of moisture per 1 B.t.u. required to be removed. This ratio determines the angle with the horizontal axis which a line, drawn through the point of 70° dry bulb temperature and 58.3° wet bulb temperature (A), will assume to fulfil the above requirements. In other words, the action must move along a line fulfilling the condition of a change of 1.56 grains of moisture for every 1 B.t.u. change.

In every case we can locate one point on the chart like A in the foregoing example. In order to draw the line showing the ratio of grains to B.t.u., it is necessary to know the location of a second point on this line. Knowing two points the line can be drawn. To find the second point the following method may be used:

$$\text{Grains at known point} - \frac{\text{Grains}}{\text{B.t.u.}} \times C = \text{Grains at unknown point,} \quad (1)$$

$$\text{B.t.u. at known point} - 1.0 \times C = \text{B.t.u. at unknown point,} \quad (2)$$

C being any convenient multiplier.

Applying this simple method to the above case we find.

$$54.4 - 1.56 \times 5 = 46.6 \text{ grains} \quad (1)$$
$$25.06 - 1.0 \times 5 = 20.06 \text{ B.t.u.} \quad (2)$$

Plotting these values locates point D, Fig. 3. The line through A and D may now be drawn. This simple rule makes it possible to locate easily the two points necessary to determine the line.

Extend this line until it intersects the saturation curve. The intersection will indicate the temperature at which the conditioning air must enter the atmosphere of the space. This point (B) shows that this temperature will be approximately 43.5°, and that it will contain 16.8 B.t.u. of heat and 41.5 grains of moisture per pound of air.

Checking back, the total heat, amounting to 25.06 B.t.u. per lb. of dry air, at 70° dry bulb and 58.3° wet bulb temperatures, less the total heat, amounting to 16.8 B.t.u. per lb. of dry air, at 43.5° saturated temperature, is 8.26 B.t.u. The moisture content, amounting to 54.4 grains per pound of dry air, at 70° dry bulb and 58.3° wet bulb temperatures, less 41.5 grains at 43.5° saturated temperature, leaves 12.9 grains of moisture removed per pound of dry air.

The ratio of 12.9 divided by 8.26 equals 1.56, which is the number of grains of moisture per B.t.u. of heat to be removed from each pound of air handled by the conditioning unit. This is the same ratio as that required by the conditions applying to the space to be cooled.

The total number of heat units to be removed per hour is 1,440,000 B.t.u. The number of total heat units removed per pound of dry air in the mixture is 8.26 B.t.u. The quotient of 1,440,000 divided by 8.26 gives 174,334 lb. of air to be passed through the conditioning apparatus each hour. Checking this weight of air for the removal of moisture, 174,334 times 12.9 grains removed per pound of air gives 2,249,000 total grains removed per hour.

Since the air leaving the conditioning apparatus is in a saturated condition, refer to the saturated volume curve, Fig. 2, where it is found that 1 lb. of dry air, saturated at 43.5°, occupies 12.795 cu. ft. of space. Therefore, the apparatus will be required to handle 2,230,600 cu. ft. of air per hour, or approximately 37,200 cu. ft. per min.

In problems involving the mixture of two bodies of air under different conditions of temperature and moisture content, the resultant condition of the mixture will lie upon a straight line joining the points representing the conditions of the two bodies. Also, the point representing the condition of the final mixture will divide that line into parts, the lengths of which will be inversely proportional to the weights of dry air comprising the two bodies.

For instance, in the example just cited, assume that it is desired to introduce the conditioning air into the space to be cooled at 60° instead of at 43.5°. Also assume that the 60° temperature is to be attained by mixing the air coming from the conditioning apparatus with a body of air coming from the space to be cooled, but not passing through the conditioning unit.

Referring to Fig. 3, the resultant condition of the final mixture will lie at the intersection C of the line AB and the 60° temperature line. The weight of air

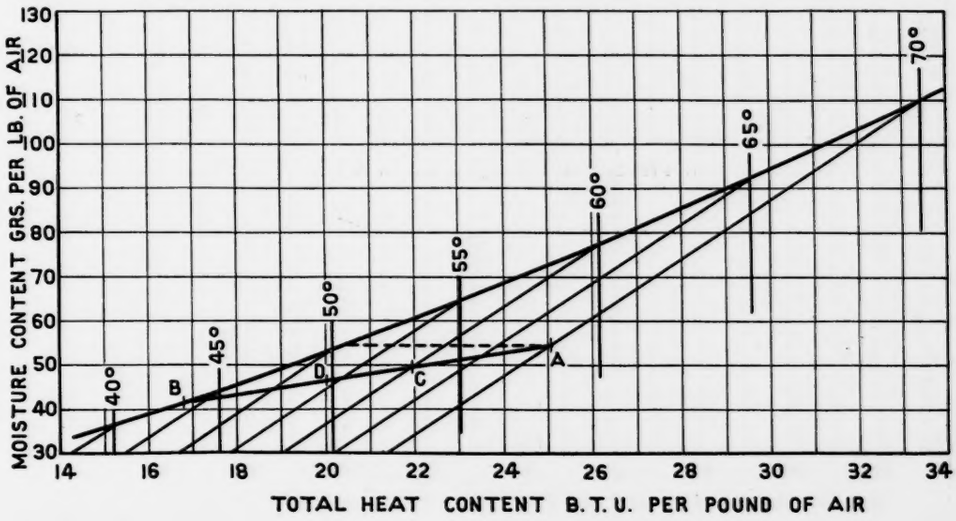


Fig. 3. Solution of Example 1

coming from the conditioning apparatus will bear that relation to the weight of air coming from the cooled space which the length of the line AC bears to the length of BC.

The lengths of these two lines are proportional to the differences between the respective total heat units of the points defining them. These total heat units are read directly upon the horizontal axis of the heat-moisture chart.

The total heats at the points A and B have already been determined to be 25.06 B.t.u. and 16.8 B.t.u. respectively, and the heat corresponding to the point C is found to be 21.95 B.t.u.

The fact that the air coming from the conditioning apparatus is mixed with a body of recirculated air, instead of going directly to the space to be cooled, has no bearing upon either the amount required or its temperature. Therefore, the same amount, 174,334 lb. per hr., will still be required to pass through the conditioning apparatus.

In order to warm this body of air, containing 41.5 grains of moisture per pound, from 43.5° to 60°, with 49.5 grains of moisture per pound, and to do this with recirculated air at 70°, containing 54.4 grains of moisture per pound, will require $174{,}344 \times \frac{21.95 - 16.8}{25.06 - 21.95} = 288{,}690$ lb. of recirculated air per hour.

The total quantity of cooling medium then going to the space to be conditioned amounts to 463,000 lb. per hr. At 60° dry bulb and 53° wet bulb temperatures, this is equivalent to 6,127,800 cu. ft. per hr., or approximately 102,130 cu. ft. per min.

Example 2

Assume an outside condition of 98° dry bulb and 79° wet bulb. For these temperatures the effective temperature will be about 86°. Let us assume an inside condition to be maintained so that the inside effective temperature will be midway between the outside effective temperature and 66°, the point of maximum comfort on the effective temperature scale.[1]

The inside effective temperature would then be 76°, with a dry bulb of 81½° and a wet bulb of 69¾°. Assume further that the theater has a seating capacity of 3000 and that the total heat input from all sources except ventilation is 1,730,000 B.t.u. per hr., and that the moisture to be removed per hour amounts to 2,700,000 grains.

* * *

On the heat-moisture chart, locate the point of intersection of the 81½° dry bulb temperature line with the 69¾° wet bulb temperature line. From this point to the saturation curve draw a line representing a ratio of $\frac{2{,}700{,}000}{1{,}730{,}000} = 1.56$ grains of moisture per B.t.u. The intersection of this line with the saturation curve indicates the temperature at which the air must leave the conditioning apparatus. This is found to be approximately 61°, with a heat content of 26.73 B.t.u. and a moisture content of 80.3 grains per pound of dry air each.

Under the conditions maintained inside the theater, the heat content is 33.27 B.t.u. and the vapor content 90.5 grains per pound of dry air. Therefore, the amount of heat absorbed by the air after leaving the conditioning apparatus will be 33.27 — 26.73 = 6.54 B.t.u. per lb. of dry air, and the moisture picked up will be 90.5 — 80.3 = 10.2 grains per pound of air.

The total amount of air required to be passed through the conditioning apparatus is determined by dividing

[1] The writer suggests that this seems to be a practice which should be more generally followed in air conditioning design. The inside air condition should vary with the outside condition, but to maintain a constant temperature difference causes discomfort when the occupant leaves the conditioned space. By varying the inside condition through *half* the range, the writer feels that the condition would be more healthful and furthermore reduce to a marked extent the power costs for the conditioning equipment.

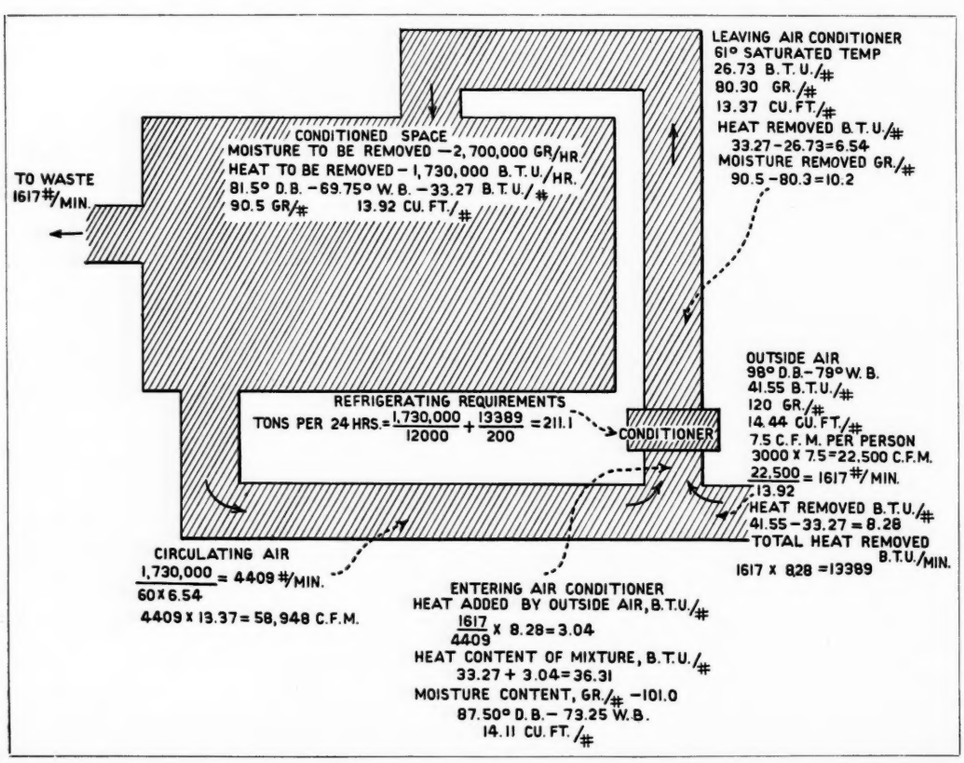


Fig. 4. Final result of solution for Example 2

the total heat, 1,730,000 B.t.u., by 6.54, which amounts to 264,526 lb. per hr. or 4409 lb. per min.

To check for the total amount of moisture removed, multiply 264,526 by 10.2 and the result is 2,698,165 grains.

For determining the total refrigerating capacity required, the heat to be removed from the ventilating air, in cooling from the outside conditions down to the conditions inside the theater, must be added to that removed from the theater itself.

If a line is drawn on the chart connecting the point representing the outside conditions to that point representing the inside conditions, the resultant of the mixture will lie somewhere on that line.

Assuming that 7.5 cu. ft. of outside air are required per minute for each person present, since the specific volume of the air at the assumed inside conditions is 13.92 cu. ft. per lb., the total weight of outside air required amounts to $\frac{3000 \times 7.5}{13.92} = 1617$ lb. of air per minute.

It has already been determined that the total weight passing through the conditioning apparatus is 4409 lb. per min. Since the ventilating air is mixed with the return air from the theater, this ventilating air forms $\frac{1617}{4409} = 36.6\%$ of the air going to the conditioning apparatus.

The total heat content due to the outside atmospheric conditions amounts to 41.55 B.t.u. per lb. of air. The difference amounting to 8.28 B.t.u. per lb. of air between this heat content and that within the theater is the heat abstracted per pound from the ventilating air. The total weight of ventilating air, 1617 lb. per min., multiplied by 8.28, is the total refrigeration required for ventilating, or 803,325 B.t.u. per hr. This is at the rate of 66.9 tons per 24 hr.

Adding the ventilating requirements to the theater load, the total refrigeration required, under the above conditions, amounts to 2,533,325 B.t.u. per hr., which is at the rate of 211.1 tons per 24 hr.

The properties of the air going to the conditioning apparatus may be determined from the line which connects the points on the chart representing the outside (98° dry bulb, 79° wet bulb) and inside (81.5° dry bulb, 69¾° wet bulb) conditions. The length of this line projected down to the horizontal axis is 8.28, *measured in B.t.u.* The lengths of the portions of this line, corresponding to the ventilating air and the return air from the theater, will be inversely proportional to their respective weights. If 8.28 represents the total air going to the conditioning apparatus, $\frac{1617}{4409} \times 8.28 =$ 3.04, is the equivalent of the ventilating air.

Since these values are inversely proportional, the length equivalent to 3.04 is adjacent to the point of inside temperature conditions. Therefore, the addition of 3.04 to 33.27, or 36.31, is the total heat content per pound of air going to the conditioning apparatus. Also its moisture content is 101 grains per pound, its dry bulb temperature is 88°, and its wet bulb temperature, 73¼°.

As a check for the above figures, the heat content per pound of air going to the conditioning apparatus is 36.31 B.t.u. and upon leaving the heat content is 26.73 B.t.u. per lb., a difference of 9.58 B.t.u. The total weight of air handled is 4409 lb. per min. Therefore, the total heat abstracted in the conditioning apparatus amounts to 42,238 B.t.u. per min., or 2,534,293 B.t.u. per hr., which is the sum of the heat from all sources, including ventilation entering the theater.

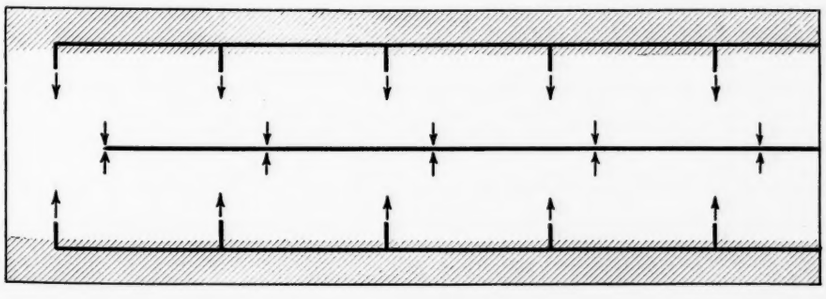
From the saturated volume curve, the specific volume of the air leaving the washer is found to be 13.37 cu. ft. per lb. Hence, if there is no recirculated air mixed with the conditioned air, the capacity of the circulating fan must be 58,948 c.f.m.

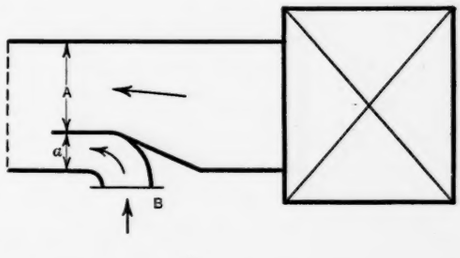
Fig. 4 shows the data for the foregoing problem in summarized form. It should be noted, however, that these calculations refer only to that portion of the air passing through the conditioning unit, no account being taken of air which may be recirculated without passing through the air conditioner.

Ventilating Dead Space in a Basement

WHEN installing the ventilating system in a basement of a recently completed office building, an attempt was made to eliminate the usual dead space back of the supply ducts. The sketch shows the plan of a room with two supply ducts along the walls and a single exhaust duct in the center. The space along the wall, shown in shaded lines, was more or less "dead."

Each of the supply outlets of this system was equipped with an ejector arrangement, a sketch of which is shown in the drawing, left. Air is drawn from the space B by the air forced from the duct through the area A. The ratio of the area A : a was 2 : 1.





Making the Rounds with a SERVICE *Man*

Part V

By
V. D. Milliken

Doc and Mac ascended to the lobby directly above the pump location and listened. "There it goes," whispered Mac. "It seems more noisy up here than it does in the boiler room."

Doc White is the service engineer for a large heating contractor. He recently engaged an assistant, McKay, who, having had no previous training in heating, is anxious to learn about the principles of vacuum heating and the methods used in correcting trouble jobs.

Each installment of this series is complete in itself.

* * *

Doc and Mac next visited a job where the complaint was that the vacuum pump was noisy.

"Gosh, Doc," said Mac, "that pump doesn't seem to be a bit more noisy than any of the other pumps we have heard running on the other jobs."

"It isn't as noisy as some of them," replied Doc, "but it is installed in a different place and in a different

manner than the other pumps. Here they have the pump installed in the corner on a concrete foundation which is built against the outer wall and the partition which separates the boiler room from the laundry room. The outside wall is concrete and the partition is of hollow tile."

Doc glanced around the room at the return and discharge piping to and from the vacuum pump. With hands on his hips and nodding his head, he sighed. "It's a shame that such a nice piece of workmanship as this has to have something wrong with it. Just look at the way those pipes leading to and from the pump have been cemented into the hollow tile walls. It makes a nice looking job, but any vibration in the pipe line is transmitted to the wall which acts as a sounding board. Such noise is readily transmitted through the building construction which consists chiefly of hollow tile and concrete. Now take a look at that pump foundation. It is bad enough to have a piece of moving machinery installed near a wall, but you will note on this job that the concrete foundation has been poured so that the concrete wall and the hollow tile wall form two of its sides. It is, therefore, practically a part of the building walls and of course will transmit any vibration imparted to it."

"What can be done to eliminate the noise?" asked Mac. "I gather from what you say that the noise is transmitted through the walls of the building rather than through the air."

"That is correct. Practically all of the noise which is heard upstairs is, I believe, conducted through the building material."

"If that is the case, then we must separate the pump from the foundation so that the vibration will be eliminated. Cork is good for that purpose isn't it, Doc?"

"Yes, cork is an excellent material for isolation. Unfortunately, however, the application of cork as an isolator is not thoroughly understood. What would you do, Mac, if you were told to put a cork foundation under this pump?"

"I would get a big mat of cork about the size of the concrete foundation and about two inches thick and put it under the pump to separate the pump from the concrete," replied Mac.

"That is just the manner in which the majority of cork pads are installed, and that is why I said that the application of cork as an isolator was not generally understood. I'll give you an example which I believe will bring to light the point I wish to stress. You have noticed the large elliptical leaf springs used on locomotives, haven't you?"

Mac nodded in the affirmative.

"Well," continued Doc, "suppose you were to take four of those immense springs from the locomotive and install them on your automobile in place of the regular springs. What would happen?"

Mac laughed as he said, "The old bus would ride more like a lumber wagon than it does now."

"Why?" was Doc's question.

"Because the springs are too stiff for such a relatively light load," replied Mac.

"Exactly. It is a question of load, and unless there is sufficient load to give the proper compression, the effect of the spring is nil. It is exactly the same with cork. Cork is composed of thousands of minute air cells confined within its structure. These air cells, when sufficiently compressed, act as the springs on your car. If you don't have sufficient compression, however, you get the lumber wagon effect. Take this pump and put cork under the whole base and you would have approximately 20 inches $\times$ 40 inches, or 800 square inches of cork. The weight of this pump is approximately 1300 pounds. Divide the 800 into 1300 and you get a load of approximately 1.6 pounds per square inch of cork. In order to be effective, the load should be about 35 pounds per square inch or more. Now, figuring the amount of cork which would give the proper load for this pump, we take the weight of the pump, which is approximately 1300 pounds, and divide it by 35 pounds, which represents the minimum load per square inch. This gives us about 37 square inches of cork instead of 800 square inches. Do you see now why a cork pad under the whole pump would be useless?"

"I sure do," replied Mac, "but how are you going to get such a small quantity of cork under the pump?"

"The best way to do it is to put a small pad of cork under each of the corners of the pump. We figured that approximately 37 square inches of cork would be sufficient to give the proper compression. If we put a pad under each corner, the pad will be one-fourth of 37 or approximately nine square inches. A pad three inches square is rather small, but if it is natural cork, bound together with a band of iron, such as may be purchased on the market for the purpose, it will do very nicely. Four of these pads, each three inches square, will do the trick. Such pads must be used with care, however, because some makes of pumps are mounted on bases which, if sprung, might cause misalignment or rubbing of the rotating parts due to close clearances. In general, pumps which have cast iron box construction bases and casings cannot be distorted due to an uneven foundation; therefore, cork pads may be used without caution, other than to have the weight of the pump distributed equally on the pads."

Mac had been listening with interest and said, "It certainly is surprising to note the small amount of cork required."

"As Al Jolson might say, 'you ain't heard nothin' yet,' because I haven't told you about rubber," said Doc. "You see, when you use cork pads which are bound with strips of iron you must elevate the pump more than one inch because of the thickness of the pads. This is a simple matter when the pumps are first installed, but on an existing installation like this one it is rather difficult to elevate the pumps without disturbing the piping. However, you have comparatively no difficulty in jacking up the pump to a height sufficient to slip four pads of one-half inch compression rubber underneath, and the nice part of it is that the same size pads will give excellent results. In other words, a load of approximately 35 pounds per square inch of area gives sufficient compression to give excellent results. I have some rubber pads with me which are three inches square. Let's go upstairs and listen to the noise complained of, then we will isolate

the pump and again return to the upper floor and listen."

Accordingly, Doc and Mac ascended to the lobby directly above the pump location and listened. Presently the pump started on automatic control.

"There it goes," whispered Mac.

"Now put your ear against that wall which is directly over the pump," said Doc.

"Holy smokes!" exclaimed Mac. "It seems more noisy up here than it does in the boiler rooms."

At this moment, the manager of the building appeared and after a short conversation they learned that the noise was heard on the second and third floors immediately above the boiler room. The manager consented to the changes which Doc suggested to remedy the transmission of noise and Doc and Mac returned to the boiler room.

They elevated the pump with a crowbar and put one of the rubber pads under each corner of the base of the pump which was flat. Then Doc instructed Mac to take a hammer and chisel and chip away all the cement which had been so carefully troweled around the pipes which passed through the hollow tile walls so that there no longer was contact between the pipe and the wall. Doc busied himself by placing strips of thin rubber between the pipe and pipe hangers so that vibration could not be transmitted through the hangers to the concrete ceiling from which they were suspended. When completed, they returned again to the lobby.

"Now, Mac, you stand in the center of the lobby and tell me when the pump starts. I will place my ear on the wall and listen for it to start."

After a short wait Doc straightened up and smiled at Mac. "Didn't hear it, did you? Put your ear against that wall now."

"I can hardly hear it now," said Mac in surprise. "That isolation is sure swell. Don't they cut the pipes, sometimes, and insert pieces of rubber hose to stop transmission of noise?"

"Yes," replied Doc, "that is frequently done, but you have noticed the wonderful improvement which we have accomplished without cutting the pipe and using rubber hose. I believe that most of the vibration is transmitted through the building construction rather than through the pipes. A common mistake made when installing rubber hose connections is to install the hose in the pipe at a point where it is likely to be filled with water. Water is an excellent conductor of vibration; therefore, if it bridges the section of hose in the pipe, it transmits vibration past the hose."

"When you find a pump which is actually noisy, what is generally the cause of the noise?" asked Mac.

"There are several conditions which cause noise, and all of them are brought about by either negligence or lack of knowledge regarding the proper operation of a vacuum pump. Some vacuum pumps are not intended for high vacuum service, and therefore if any adjustments is made to create a high vacuum, the pump may become noisy. The operators of vacuum pumps frequently tamper with the vacuum breakers or relief valves, and in so doing raise the original setting, thus producing a high vacuum. Some pumps rumble while other pumps whine. Too great a pressure is often the cause of noise. If the total head at the pump exceeds that for which the pump is designed, it may cause noise. In other words, noise is caused by abnormal operation, and if a pump is noisy, determine the cause before condemning the pump. Ball bearing failure is another source of noise. Ball bearings should give years of service. Unfortunately, they are sometimes poorly lubricated. Cheap greases contain fatty acids which attack the balls and races of the ball bearings. Good grease is cheaper than ball bearings. If the bearing runs dry, or becomes pitted, naturally it will be noisy. Misalignment of the pump and motor shafts will cause noise. A straightedge and pair of calipers are generally sufficient to check alignment. Another common cause of noise is steam entering the pump, or condensate at so great a temperature that it flashes into steam."

"Well, Mac, that cleans up all our trouble jobs for the present. Let's go back to the office and I'll go over a few miscellaneous points which we haven't covered so far."

[*To be concluded*]

NEXT MONTH!

The October number of HEATING AND VENTILATING will be largely devoted to the Air Conditioning of Small Buildings, including residences, small stores and similar establishments.

The editorial staff has been working on this special reference issue for many months and has had the cooperation of manufacturers, engineers and contractors in preparing the material to be presented. It will be by far the most complete analysis of the subject and classification of the available apparatus so far published.

Every engineer, contractor and architect concerned with the growing demand for air conditioning will want the October issue of HEATING AND VENTILATING for study and reference.

EDITORIALS

The 1932 Building Market

ONE of the year's surprises has been the persistent and unchecked falling off in the volume of residential building. Last year was a bad one for residential construction but many felt that the accumulating deficit of residential space would surely be reflected in this year's totals.

This has not worked out, and now come the Department of Labor's July records with some startling figures. Residential building permits during July of this year were 80.5% less in dollar value than July of last year, while in non-residential buildings the figure was 64.8% less. In number of buildings residential buildings fell off 67.6% against 45.9% for non-residential buildings. This year is one in which non-residential building continues to dominate the building industry.

Contractor Licensing

FOLLOWING on the heels of the adoption of a contractors' licensing plan in California and a movement for a law in New York during the past winter, agitation is now being carried on for such a plan in Ohio. As in the other cases, the project would include heating contractors in its scope. The comment we receive is to the effect that the proposal is not likely to make much headway in Ohio this year.

This whole movement looking toward some kind of licensing plan is of vital interest to all present and prospective contractors. Its adherents look on the idea as the salvation of contracting, while those opposed point to alleged failures of other like plans. Just now these proposals are all in the discussion stage except in California.

One general principle may well guide the thinking of any contractor who must make up his mind whether or not to support such moves. It is, that the success of licensing laws has depended almost altogether on the character of the agency set up to enforce them. No matter how well the law itself may be drawn, or how promising the plan, unless the actual administration is free of politics—even group or industry politics—it will not succeed for long. Before supporting such proposals, make sure that their administration will be in competent hands. Where this is true we believe that licensing plans will work to advantage.

Building and Costs

MANY have assumed that credit is the one factor which is holding back building activity. Dr. H. Parker Willis, formerly secretary of the Federal Reserve Board, does not agree with this view. Writing in the New York *World-Telegram* he states that the cessation of not only small home building but the building of nearly everything else is "found in the fact that it is, for practical purposes, pretty generally impossible to build a small or moderate-sized house that will when completed be worth what it cost." He argues that costs of labor, of transportation and of construction management must be reduced, and that there will be no upturn in building until these are effected no matter how much credit is made available.

To those of us who know full well that high building costs are not due to excessive profits from materials and apparatus, this statement by the doctor is confusing, at least at first thought.

How can a building not be worth its cost? Evidently he does not refer to the structure itself but to relative costs or to the value of the satisfaction obtained from ownership. An example of relative value is: If the cost of a building is five times as many dollars as the cost of an automobile, can I get as much value in satisfaction and pleasure out of one building as out of five cars?

The unfortunate feature of building construction has been that when weighed on such a scale of relative values it has really offered its owner but little in the way of measurable satisfaction. He has always obtained shelter—and little else. He has gotten little of the thrill which goes with the performance of a new car—few features to call to the envious attention of admiring friends. The sooner all of us get to realize the values which may be attached to those features which give satisfaction the sooner more of them will be incorporated and the sooner criticisms like those of Doctor Willis will no longer apply.

Fortunately, we may say that no other equipment of the modern building has shown such strides toward offering satisfaction features at little or no increase in cost in the last few years as have the heating plants. Unfortunately, we are not yet able to secure their installation in all buildings. Unfortunately, too, we have little or no control over transportation and labor costs of the building as a whole.

Drafting Symbols

EVERYONE who has drafting to do or to supervise will be interested in the recently published proposal regarding uniform symbols for this work prepared by a committee of the American Standards Association. The heating and ventilating, pipe and fittings, and heat-power symbols combine to take up a large part of the pamphlet. The report is presented for comment and criticism and now is the time to make them. Copies of the report are available to those interested.

In general, the symbols proposed are similar to those in use, but in many cases changes are made which tend to simplify existing symbols. The heating and ventilating symbols show rather markedly the effect of introducing some of the German symbols, while combining them with symbols on piping, heat-power, electrical apparatus and plumbing gives comprehensiveness to the work of the committee.

Abstracts . . .

Reviews and Summaries of . . . . Current Papers, Books and Pamphlets

Tests of Convectors in a Warm Wall Testing Booth

By A. P. Kratz, M. K. Fahnestock, and E. L. Broderick

Paper presented at the Milwaukee meeting of the A.S.H.V.E., June, 1932; 14 curves and graphs, two tables; approximately 4000 words.

A study of the differences in test results produced by testing convectors in a warm wall booth and in a cold walled room, and of the limits of application of the Dietz equation to convectors.

Following conclusions:

(1) The correction factor $C = \left(\frac{150}{t_s - t_1}\right)^{1.3}$ for reducing the heat output of a convector under test conditions to the heat output under standard conditions, with steam temperature of 215° and inlet air temperature of 65°, is applicable within a probable error of 1% for temperatures of the air at the inlet between 60° and 80°, and for overall heights of from 25 in. to 32 in. with a steam temperature of approximately 215°.

(2) For overall heights outside of the range of 25 in. to 32 in. it is possible that the error resulting from the application of the correction factor does not exceed 5.0%.

(3) The heat output of a convector under service conditions in an actual room will be within 10% of the heat output determined from tests in a warm wall booth, with the same temperature for the steam and for the air at the inlet in both cases, provided that the size of the convector is sufficient to heat the actual room to a temperature of approximately 68° at the 30-in. level.

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Thermal Properties of Building Materials

By F. B. Rowley and A. B. Algren

Paper resulting from research conducted at the University of Minnesota with the A.S.H.V.E. co-operating; presented at the Milwaukee meeting, June, 1932.

Reports tests made on various building wall constructions in terms of the coefficient U, or the overall transmission coefficient. Some of the data resulted from tests made by the hot box method, while in other cases results were calculated from coefficients previously obtained and checked against the overall test figures. Agreement of the test results with the computed results are reported in some cases.

The tests reported covered various constructions using clay tile, concrete, brick, and rubble walls. A table is included in which the coefficients for these walls are listed.

As data were at hand obtained under several mean temperature conditions, the variation of the coefficients with the mean temperature was plotted for a number of wall constructions. The curves show that, as would be expected, the coefficients increase as the temperature increases. The slope of the lines is much the same for all the wall constructions reported except one in which aluminum foil was used. In this one the change in the coefficient with change in mean temperature was very slight over the range from 40° to 110°.

When surface resistances and the resistance of air spaces were examined in some of the walls tested, the results showed clearly that the use of aluminum foil on the sides of these spaces greatly increased the resistance of the air space to the floor of heat. This result is in line with those reported by other investigators who have demonstrated the importance of radiant heat as a factor in heat transmission through walls.

Other confirmatory evidence brought out by the authors' tests showed that the heat conductivity of concrete walls decreases with age and that it is better to use tile which break up the heat flow path rather than those shapes which give a more direct path.

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The Flow of Air Through Circular Orifices in Thin Plates

By Joseph A. Polson and Joseph G. Lowther

Bulletin No. 240 of the Engineering Experiment Station of the University of Illinois; illustrated and tabular; 6 in. x 9 in.; 40 pages; obtainable by addressing the station at Urbana, Ill.; price 25 cents.

The authors describe their testing method which consisted of the use of weighing tanks for finding the quantity of air passing through the orifices, and present and analyze their results. Among the conclusions they reached are the following:

1. The coefficients of discharge through square-edged orifices in thin plates increase in value with an increase in inlet pressure up to the critical pressure, when the coefficient is based on Durley's equations; that is the coefficient is the ratio of the weight of air actually discharged to the weight that would be discharged when computed by Durley's equation.
2. The plotted values of the coefficients for orifices

of various diameters show that all have the same general slope, increasing with increased inlet pressure. Within the range of orifice diameters and pressures used in these tests, all coefficients might be represented by a single line such that its values would be within 1% of the values actually found. For work in which a tolerance of 1% is permissible this arrangement would simplify the calculations.

3. Square-edged orifices should be very carefully used and preserved. Very slight injury to the entrance edge may cause a change in the coefficient of from 2% to 3%, which may not be detected until the orifice is recalibrated.

●

Effect of Sun's Heat on Fuel Consumption

From "Die Warmestrahlung der Sonne und Ihr Einfluss auf den Brennstoffverbrauch von Heizungsanlagen," by A. Eigenmann; Gesundheits-Ingenieur, Berlin, No. 10, March, 1932; approximately 2500 words.

ABSTRACTED BY A. A. BERESTNEFF

Measurements are being made in several European meterological stations to determine the heating effect of sun radiation on surfaces in various positions. The accompanying table contains data recorded at Arosa.

HEAT ABSORBED PER DAY BY A CUBE, AVERAGE PERIODS OF SUNSHINE, B.t.u. per SQ. IN.

		Horizontal	South	East	West	North	Total
January	15	1420	3930	900	750		7000
February	15	2590	4870	1430	1380		10270
March	15	3680	4360	1940	1820		11800
April	15	4500	2980	2310	1970	16	11776
May	15	4540	1740	2410	1720	179	10589
June	15	4980	1400	2560	1770	326	11036
July	15	5700	1820	2900	2100	326	12846
August	15	5820	2980	2980	2500	98	14358
September	15	4700	4070	2230	2080		13080
October	15	2860	4400	1460	1460		10180
November	15	1820	4440	1060	1020		8340
December	15	1190	3690	780	570		6230
Total for the year in 1000 B.t.u. per sq. in.		1330	1233	700	583	27.6	38800

These data are useful to engineers in determining the heat quantities from the sun affecting fuel consumption in heating plants, capacity and energy consumption of air conditioning plants and similar problems. However, as the writer states, there is not sufficient information available for making a correct estimate of the percentage of the radiated heat retained. He therefore recommends a continuation of the investigation to determine definite values for this percentage.

●

The Calculation of Heat Transmission

By Margaret Fishenden and Owen A. Saunders

Cloth-bound book; 6 in. x 9 in.; published by the Stationery Office, London; obtainable from the British Library of Information, 270 Madison Ave., New York; $2.30 postpaid; 280 pages.

The book is in the nature of a review of the existing experimental data on the subject. The authors state the principles of heat transmission by the three methods and then quote the findings of others and analyze and compare them. The result is that they have assembled together an outline of the existing information and have put much of it into the book in enough detail to permit its use. Examples of calculations for a number of cases are given and these, being in English units, are easily followed. One of the difficulties encountered by the average American engineer who attempts to follow through experimental work on heat transmission is that so much of it has been carried on by continental experimenters who have reported their findings in their own languages. Consequently, the assembly of this information into a usable English form is welcome. For those who wish to go to the sources, a bibliography of 198 titles is listed.

●

Automatic Gas Burners

By C. George Segeler

Paper presented at the Milwaukee meeting of the A.S.H.V.E., June, 1932; one table, five curves and graphs; approximately 4000 words.

Outlines the nature of automatic, or conversion, gas burners and describes the three types in use, together with a discussion of installation methods and an explanation of results obtained by analyzing operating data from 700 installations. By the method used by the author in analyzing these figures, he shows that the gas-designed installations were operating at an average efficiency 12% higher than that obtained with converted installations. He also notes in passing that the calculated heat loss data used were scarcely as accurate as might be wished.

●

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage

By F. C. Houghten and Paul McDermott

Paper presented at the Milwaukee meeting of the A.S.H.V.E., June, 1932; illustrated and with two tables of results; approximately 4000 words.

The authors made studies in a garage in Pittsburgh which was equipped for window ventilation. They set up propeller fans in window openings in order to control ventilation air. Surveys were made in this garage to determine the amount of air necessary to keep carbon monoxide concentration low with varying car operations. Eight surveys were made. From the results obtained, curves were plotted and a table prepared. The studies are not conclusive but the high concentrations of carbon monoxide were found at the ceiling, and it seems that air supply at the floor and exhaust at the ceiling is the preferable arrangement. The data of the authors indicate that 29.4 cu. ft. per hr. per car for cross ventilation should be allowed, and 31.7 cu. ft. per hr. per car for upward ventilation. These values are lower than those obtained some years ago by other investigators studying a somewhat similar problem.

Your Everyday Problems

Mr. Hanburger will be glad to answer your problems relating to installation and operation of heating and ventilating systems, and wherever possible will give the reasons for his suggestions. Letters will be answered direct to the inquirer and in some cases published anonymously. If sketches are included they should be clearly drawn. Write Mr. Hanburger, HEATING AND VENTILATING, 148 Lafayette Street, New York.

Conducted by FRED W. HANBURGER

QUESTION 54. I have used concealed radiators on a hot water installation, but all throughout the house it appears as though the amount of radiation is not sufficient. I have used the same radiator on steam and employed the same method for calculating my heat loss with entire satisfaction. Is there any reason for this due to the radiator in its connection with hot water?
M. R.

ANSWER. In a concealed radiator much of its efficiency depends upon the velocity of air due to the stack effect of the covering, the rate of emission varying roughly as the square root of the velocity, especially when low, such as prevail in this type of radiator. It is, therefore, only natural to suppose that when such radiators are used in a hot water system, where the temperature of the water is considerably less than steam temperature, the air flow is reduced and so will be reduced the emissivity per unit of area of the heating medium. Some of the manufacturers of these radiators can give you a true rate of emission for their radiators for hot water use. In some cases you will find this to be as low as 140 B.t.u. per sq. ft. of their equivalent area, instead of the generally accepted value of 160 B.t.u.

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QUESTION 55. I am a student, and having access to your magazine I find lots of interesting things in it.

In your June issue, Question 48, is a very interesting discussion on orifices. The method of analysis is plain. Now, won't you complete it and write me the size of the orifice which you calculated and how this size is fixed. I'll appreciate it. T. L.

ANSWER. While it may sometimes be uninteresting to some of the other readers to refer to a back question, I have received several other inquiries on this same question and, therefore, feel that it may be of more or less general interest.

The method of calculating the area for this depends upon the velocity that was necessary to create the proper amount of friction through the orifice; namely, 0.13 in. of water. To reduce this to velocity, we first find its pressure in pounds per unit of area. Since 1 in. of water is equal to 5.2 lb. per sq. ft., 0.13 in. would create a pressure of 0.676 lb. per sq. ft.

We can now obtain the head in 70° air by dividing this figure by the weight of 1 cu. ft. of this air, which is 0.075 lb., giving us 9 ft.

The velocity per second then is obtained by the formula $V = 8\sqrt{H}$ so that $V = 8\sqrt{9} = 24$ ft. per sec., or 1440 ft. per min.

As the original velocity through 2 ft. diameter duct was 1200 ft. per min., the area of the orifice will be $\frac{1200}{1440} \times 3.14 = 2.6$ sq. ft.

•

QUESTION 56. During the latter part of the winter I installed a pneumatic tank on the pumping equipment together with a septic tank and tile beds in a suburban home. The air becomes exhausted in the tank and it is necessary occasionally to pump in a supply with a bicycle pump. The septic tank has been in use for four weeks but there still is a quite disagreeable odor. I installed it during winter weather but had the tile laid below the frost line. B. R. C.

ANSWER. While these questions do not pertain to heating or ventilating, they are somewhat allied and I am glad if I may be of aid in the matter.

First: All pumps of this nature are supplied with a small air inlet valve whose purpose is to supply sufficient air to replace that absorbed by the water, keeping the water line always at approximately the same height. It is possible that this air valve is not functioning properly, or again, that you may have installed a larger tank than the valve can accommodate. The latter trouble may be overcome by incorporating an air cock on the suction line near the pump and setting it manually to admit just the proper amount of air.

Second: It may have been well to have seeded your septic tank before starting operation by introducing a small quantity of rotting manure to serve as yeast to start the action. A yeast cake would also serve the purpose. If this were the cause of the trouble it is very likely that the annoyance would be passed by this time as the tank would probably have become germinated from some other source.

I am afraid, however, that you have the tile laid too deeply. While it is so buried and should not give a perceptible odor, it is possible that it is oozing through the ground somewhere and when it does it most certainly will give off a disagreeable odor after being buried so deeply where no air can reach it. Air is very necessary in the final reduction, and the tile beds should not be more than about 18 in. below the surface.

There is no danger of the effluent freezing, since there still is a decomposition going on as the organic matter unites with the oxygen, and this decomposition or combustion releases heat.

It is doubtful if the temperature of the sewage will ever fall much below 50° F.

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QUESTION 57. For the past two heating seasons considerable trouble has been encountered with the heating system in the dwelling shown on the sketch. The back rooms of the house do not heat properly and it is necessary at times to carry 5 lb. to 6 lb. pressure in order to get heat in these rooms.

Will you kindly look over the sketch and advise me if the changes enumerated below will give better results.

(a) I contemplate changing the return lines so as to drop them directly from main steam lines to the floor and then back to the boiler. The return plugs are 13 in. from floor. From the 12 ft. portion on main steam pipe to the end of cellar wall is about 3 ft. and I was wondering if it would be possible to continue return line from main steam pipe out 3 ft. so that the pipe could be run down the wall and back to the boiler.

(b) There is at the present time one 2 in. steam main, but there is another 2 in. plugged connection on top of boiler. Would it be satisfactory for me to plug the tee connection on the line and then use an elbow for the second connection and have two independent lines, or would it be better to make a common header?

(c) If changes are made would an equalizer pipe be necessary?

(d) Will the changes as suggested have a tendency to eliminate water hammer?

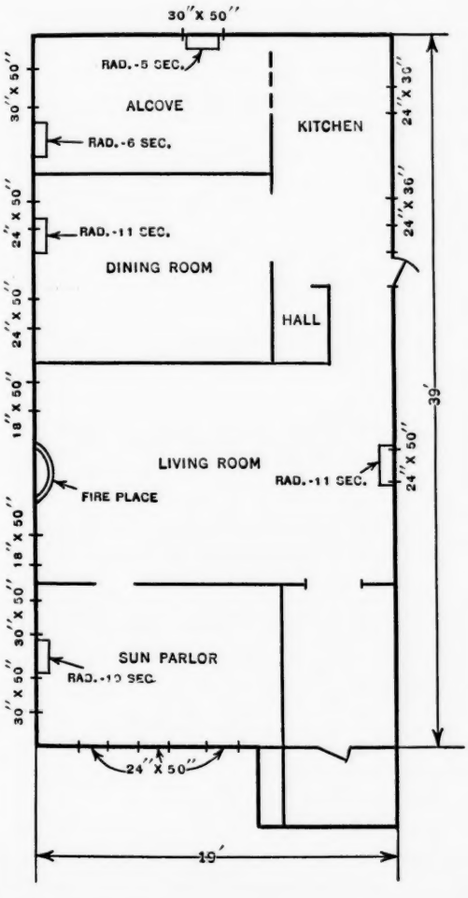
(e) Will you let me know if any of the rooms have insufficient radiator sections or if the boiler is too small to take care of any additional sections. The boiler is rated at 290 sq. ft. direct cast iron radiation.

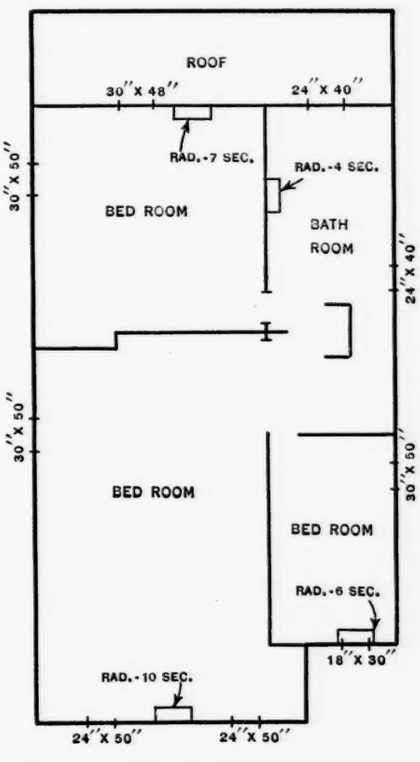
(f) I am also enclosing a sketch of the layout of the house which has 70 radiator sections, as shown in the sketch, and ask you to advise me how to figure radiation surface. The 2, 20, 200 rule does not appear to be applicable to this house. *J. J. T.*

ANSWER. I do not feel that your trouble lies in the cellar mains. From the fact that you speak of water hammer, it would appear that the branches from the mains to the risers are giving most of the trouble, especially if this is a one-pipe system. The branches to the foot of the risers should have a pitch of at least 1 in. in 5 ft., and 1¼ in. pipe should not carry more than 55 sq. ft. of radiation; 1½ in. 81 ft., and 2 in. pipe 165 sq. ft. of cast iron radiation.

The kitchen and alcove and also the second floor rooms have not sufficient radiation and it appears as though the heat losses through the ceiling were neglected in estimating their heat loss. Each of these rooms could stand from two to three sections additional radiation.

Answering (f), there are many short rules and while none will apply to all cases, due to the fact that the thickness of the walls varies in different buildings, yet they do give sufficiently close results to make them practical. Most of the rules are based on the assumption that the heat loss per square foot of wall is 20 B.t.u. and 100 B.t.u. for each square foot of glass. Thus, dividing the wall by 5 will reduce it to an equivalent glass surface; both can then be added, multiplied by 100 to give the heat loss. To this must be added the amount for infiltration, and if we accept this as approximately one air change per hour and that each cubic foot requires 1 B.t.u., we can then add the cubical contents of room to each wall and glass losses and divide the total by 240 to obtain the square feet of radiation. Where there is a roof loss the emission through the ceiling is about 10 B.t.u., which would give precisely the same results as the cubic content factor with a ceiling height of 10 ft. All this can be reduced to a simple rule expressed as follows:





0.08 × wall + 0.4 × glass + 0.004 × cu. ft. = sq. ft. C.I.R.

News of the Month

Standard Drafting Room Symbols Ready for Criticism

A proposed American recommended practice for graphical symbols to be used on drawings has been published by the American Standards Association. The work, which was sponsored by the A.S.M.E. and the Society for the Promotion of Engineering Education, is ready for criticism and comment from interested individuals or organizations.

The proposed standard covers 23 symbols for heating and ventilating drawings, 44 for plumbing fixtures, 21 for heat-power layouts, 27 for pipe and pipe fittings, and others. The symbols for heating and ventilating were taken over from the A.S.H.V.E. and partly from the German national standards in this field.

Copies of the draft are available as a loan to those interested from the American Standards Association, 29 West 39th St., New York.

University Studies Use of Filters in Relieving Hay Fever

Research investigations into the use of filters for relief of hay fever patients are being conducted by the University of Illinois College of Medicine at Chicago. As subjects for the observations, 24 men and 24 women, all victims of the malady, have been selected. These individuals are given bed and board in a pollen-free ward in the university clinic. In shifts of eight, they sleep there for seven consecutive nights. In the daytime they are free to go about as they please.

Similar experiments were carried on last year when the same number of subjects slept in the wards from August 15 until the coming of frost. It was found that while these patients slept well in the pollen-free atmosphere, they were afflicted with sneezing as soon as they breathed the outside air. The filters used last year eliminated most of the ragweed pollen, but the system installed for this year's investigations is said to completely remove the irritant from the air. Care has been exercised that this year's subjects have no other ailments complicated with the hay fever. Dr. Clarence A. Johnson, who is directing the work, said that the university does not expect to find an immediate cure as a result of its research, but hopes to develop a fairly economical and practical system by which victims may be brought temporary relief.

University of Cincinnati Offers Courses in Heating

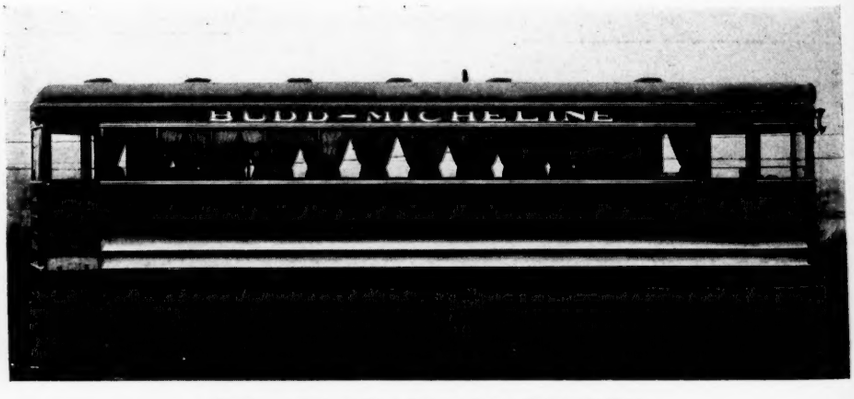
An expanded program of evening courses in commerce, engineering, and applied arts at the University of Cincinnati will offer a number of courses of direct interest to persons engaged in selling heating and piping equipment when classes commence September 26. These include psychology of selling and advertising; human relationships in business; effective speaking; combustion and steam engineering; a technical course covering fuels, combustion, heat losses, heat balance, and equipment; reasoning in business, and psychology of learning.

Registration for all evening courses in these subjects will take place in the Engineering Building on the campus September 19-23, inclusive, from 6:30 p.m. to 9:00 p.m. A copy of the new announcement may be had from the Director of Evening Courses, College of Engineering and Commerce, University of Cincinnati, Cincinnati, Ohio.

Establishment of Home Loan Banks Under Way

Organization and preliminary work necessary to put the Federal Home Loan Bank system into operation is being speeded by the Home Loan Bank Board, appointed July 5 by President Hoover. Franklin W. Fort, Newark, N. J., is chairman, and the balance of the group is composed of Nathan Adams, Dallas, Tex., William E. Best, Pittsburgh, Pa., H. Morton Bodfish, Chicago, Ill., and Dr. John M. Gries, Rosewood, Ohio.

From eight to twelve of these banks will be established, one in each of a corresponding number of districts to be created throughout the country. Each bank will be capitalized for not less than $5 million, this amount to be subscribed by institutions eligible for membership in the system, the stock remaining unsubscribed in each case to be purchased by the Treasury Department. Institutions eligible for membership include building and loan associations, cooperative banks, homestead associations, insurance companies, and savings banks which meet certain requirements.



Rubber-Tired Car Heated by Bottled Gas

Bottled gas heats this lightweight, rubber-tired, gasoline-driven rail car being tried out by the Reading Company near Philadelphia. Outside air, drawn in through the sides of the car near the front end, is warmed in compact tubular heaters in the baggage compartment before being delivered to the passenger space through ducts along the sides under the seats. The use of dry ice in the tubular heaters for summer cooling has been contemplated.

Loans may be made to three kinds of applicants which include member institutions, non-member institutions, and home owners. Non-member institutions may become borrowers only when they are otherwise eligible for membership but are prevented from purchasing stock by state law. Home owners may have borrowing privileges only when they are unable to secure mortgage loans from other sources, and this privilege will only be continued as long as the Government owns stock in the system.

Advances will be made for not more than 40% of the value of the real estate at the time of the advance, and such advances are not to exceed 60% of the unpaid principal. No home mortgage loan may be accepted as collateral security if it has more than 15 years to run to maturity, if the value of the real estate exceeds $20,000, or if the mortgage is more than six months past due.

President Vetoes Bill to Expand Wage Act

A bill to amend the Bacon-Davis law, relative to wage payments to laborers and mechanics on public works construction, was returned to the Senate without his approval by President Hoover, July 5. The bill would have required that wages paid to laborers and mechanics engaged in public building must be equal to the prevailing rates in the section in which the building is erected. It also proposed that advertised specifications for every building or public work should contain a provision stating the prevailing rate of wages.

A memorandum by William N. Doak, Secretary of Labor, which accompanied the President's veto message, indicated that the proposed amendment was unnecessary and stated that if it were to be put into operation a large investigating force would have to be set up, putting the Government to great expense, and increasing cost and inconvenience to contractors. Under the existing law controversies over prevailing wages are generally settled between contractors and organized labor, and the Department of Labor is only called upon to intervene when no decision can be reached.

July Barometer in New York Lowest on Record

July in New York was a most pleasant summer month. Showers were wanting to refreshen the greenness of vegetation which seemed burned a little at the end of the month. Only one July in 60 years had less rainfall than July, 1932. Yet the lack of water was not exhausting because the relative amount of moisture in the atmosphere was normal.

The average percentage of sunshine was 14% above normal. Storms of the extensive cyclone type, as well as tornadoes, or thunderstorms of a severe local type, were singularly absent during the month, yet no July in 60 years had a lower record of barometric pressure.

In less than a dozen other months of record, including last March, the mean monthly barometer was lower than that of July, the reading on the second, 29.35 in., being the lowest ever in July. From October, 1931, to February, 1932, inclusive, the barometric pressure was high, as was also, very strangely, the temperature. In June and July, while the mercury was low in the barometer, it rarely rose much above normal in the thermometer.

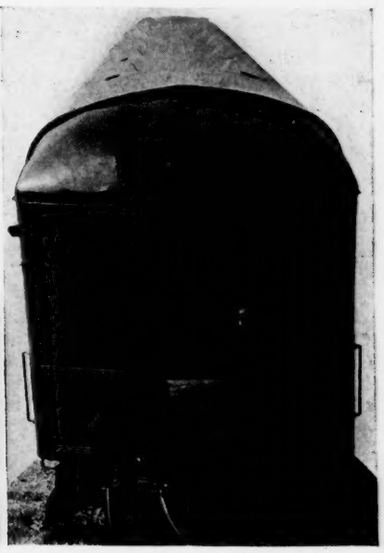
The highest temperature thus far this summer was 88°. This occurred once in June and twice in July. The average temperature for May was 0.3° above normal; that of June, 0.2° above, and of July only 0.1° above. It would be difficult to find in the records of the past three consecutive months with temperature so uniformly normal, or to conceive of them in the near future.

Modernization Campaigns Successful Throughout Country

A report received from the Home Modernizing Committee of the Department of Commerce, Washington, D. C., states that campaigns

Double-Decked Railroad Car Is Heat Insulated

This double-decked coach, placed in service on suburban lines August 13 by the Long Island Railroad, has insulating material packed between inner and outer side walls. A double roof permits free circulation of air between the two surfaces. The car is heated electrically, and has exhaust ducts in the roof. The novel seating arrangement accommodates about 60% more passengers than in a standard coach of the same size.



continue to arouse interest throughout the country in home improvements resulting in betterment of local labor conditions and stimulation of local business. Approximately $45 million has been pledged since January 1 for property repair and improvement, which will increase the employment of wage-earners during the next few months.

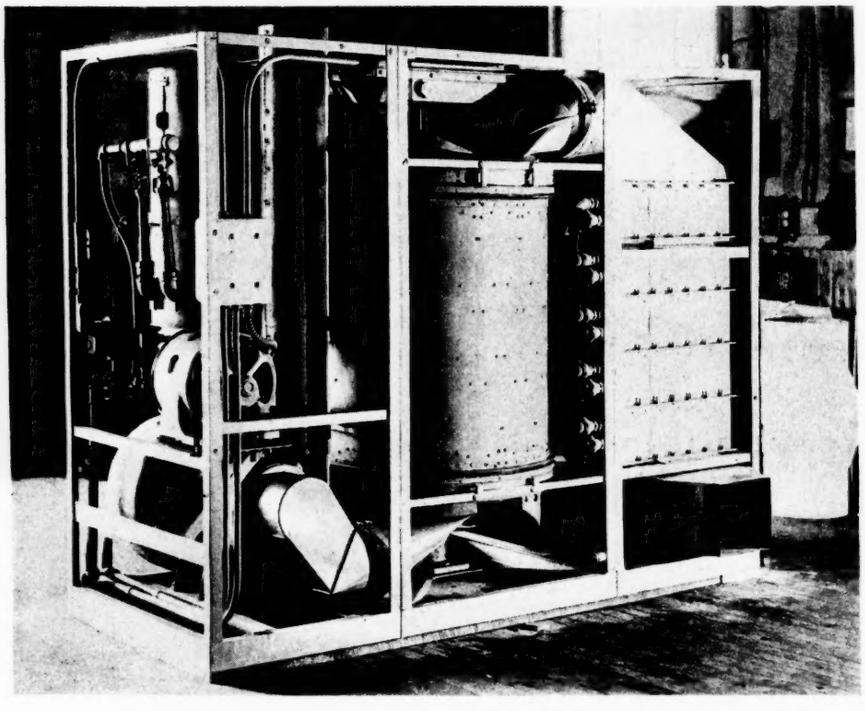
Campaigns have been most numerous in the thickly populated districts of the East and the Middle West, and highly successful results have been reported in individual cities in the Far West. More major activities have been consummated in the state of Illinois than in any other one state, with eleven campaigns now active or completed this year. Six of the cities where campaigns have been pushed have recorded pledges for home improvement aggregating $2,085,000. The states of Michigan, Ohio, Pennsylvania, New York, and New Jersey have reported nearly as many campaigns as Illinois, and the amounts spent in some cases have exceeded those in Illinois. Six cities in New York have pledged a total of $6,242,000.

In addition to the 62 cities for which data are available, there are campaigns active or completed in 69 cities for which no financial reports are at hand, and organized activities are contemplated in about 50 more.

Utility-Backed Air Conditioning Show Results in 126 Sales

An exhibition of room coolers and air conditioning equipment which opened July 11 in the main showroom of the Commonwealth Edison Co., Chicago, brought 126 sales to manufacturers of these appliances. The show was sponsored by the utility and continued for two weeks.

Appliances sold included 66 ice room coolers, 40 cooling fans, 13 humidifiers, three air filters, two furnace conditioners, and two mechanical conditioners. Twenty-two manufacturers in this field had their products on display. Prospect leads obtained at the show were referred to the manufacturers since the Commonwealth Edison Company does not make direct sales. The show was publicized by the company through extensive newspaper advertising and by direct mailing of 10,000 invitations to building managers, small store and shop owners, and others likely to be interested.



Rotating Bed Used In Gas-Fired Air Conditioner

Consolidated Gas Company of New York, Dallas Gas Company, Consolidated Gas, Electric Light & Power Company of Baltimore, and The Peoples Gas Light & Coke Co., Chicago, are cooperating in research involving the use of gas for air conditioning this summer. The illustration shows a rotary silica-gel conditioner in which a series of gel beds rotate, a certain number receiving air from which they absorb moisture. Other beds at the same time are receiving warm air from the gas furnace which reactivates the gel. Still others are being cooled by a flow of air to remove the heat of activation. The use of the rotary idea is considered a great advance over the use of the stationary beds which were employed last year. Photo, courtesy American Gas Association.

Chicago Chosen for A. O. B. A. Convention

Chicago has been selected as the meeting place of the tenth annual convention and show of the American Oil Burner Association, scheduled for the week of June 11, 1933. Convention headquarters will be at the Hotel Stevens.

Addition to Senate Building to be Air Conditioned

The contract has been awarded for a complete refrigerating and air conditioning system for the new fourth wing of the United States Senate Office Building now under construction in Washington. The specifications call for air conditioning for the entire new wing from the first to the third floor, with provision for extending the system to the basement at some future date. The contract is for $150,000.

Trade Commission Completes Investigation of Building Materials Industry

Field work that has been conducted in connection with the Federal Trade Commission's investigation of certain specific industries in the building materials field is complete, and a report covering this is now being prepared, according to the Monthly Statement of Work, released by the commission at Washington, July 22.

Under Senate Resolution 493, 71st Congress, 3rd Session, the commission was directed to investigate

and report facts relating to the letting of contracts for the construction of Government buildings, particularly with a view to determining whether there are or have been any price fixing or other agreements, understandings, or combinations of interests among individuals, partnerships, or corporations engaged in production, manufacture, or sale of building materials with respect to the prices or other terms at or under which such material will be furnished contractors or bidders for such construction work.

A.G.A. Changes Conference Dates

The dates for the second annual conference of the manufacturers' section of the American Gas Association have been changed from September 22-23 to September 29-30. The conference is to be held at the Netherland Plaza Hotel, Cincinnati, Ohio.

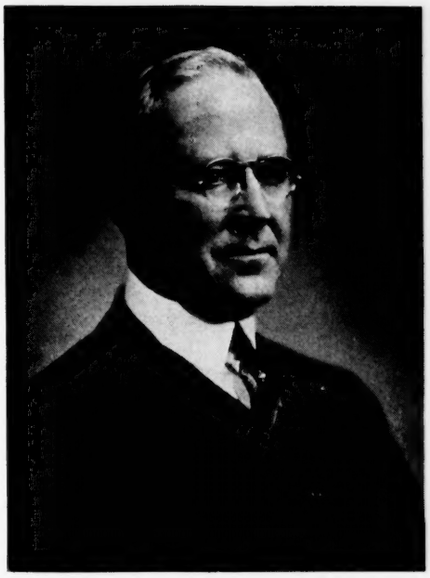
Health Association to Study Air Hygiene

Air hygiene is included among the subjects to be brought up for discussion at the sixty-first annual meeting of the American Public Health Association, which will take place in Washington, D. C., October 24-27.

Pure Air Committee Fails to Agree on Boiler Setting Heights

Standards for setting heights for heating boilers recommended by the Midwest Stoker Association and approved by the Stoker Manufacturers' Association have not been agreed upon by the sub-committee on boiler setting heights of the pure air committee of the American Society of Mechanical Engineers. The sub-committee, composed of Col. Elliott H. Whitlock, chairman, H. K. Kugel, H. B. Meller, and Charles J. McCabe, has presented its own table of setting heights for heating boilers to the pure air committee, headed by Ely C. Hutchinson.

Representatives of various interested organizations will be invited



Col. Elliott H. Whitlock, chairman of the sub-committee on boiler setting heights of the Pure Air Committee of the A.S.M.E.

by the pure air committee to attend an open meeting in which the proposed standards will be discussed. The meeting will be held in the Engineering Societies Building, New York, during September.

Federal Public Works Approaching Three-Quarter Mark

With 583 projects amounting to $388,963,542 either completed or under contract, nearly 75% of the Government building program which Congress speeded up a year and a half ago to aid employment is either finished or under way. Only 233 projects involving $107,500,400 have yet to go on the market, according to oral information made available August 6 at the Treasury Department.

COMING EVENTS

OCTOBER 3-7, 1932. Twelfth Fall Meeting of the American Welding Society, to be held in the Statler Hotel, Buffalo, N. Y.

OCTOBER 10-12, 1932. Fourteenth Annual Convention of the American Gas Association, to be held in Atlantic City, N. J.

OCTOBER 24-27, 1932. Sixty-First Annual Conference of the American Public Health Association, to be held in Washington, D. C.

NOVEMBER 30-DECEMBER 4, 1932. Annual Meeting of the American Society of Mechanical Engineers, to be held in the Engineering Societies Building, New York.

DECEMBER 5-10, 1932. Tenth National Exposition of Power and Mechanical Engineering, to be held at the Grand Central Palace, New York.

DECEMBER 7-10, 1932. Twenty-Eighth Annual Meeting of the American Society of Refrigerating Engineers, to be held in New York.

JUNE 12-17, 1933. Tenth Annual Convention of the American Oil Burner Association, to be held in Chicago. Headquarters at the Hotel Stevens.

"Fuel Oil" Publishes Survey of Oil Burner Industry

Fuel Oil Journal has published the "1932 Survey of the Oil Burner Industry," a 36-page study of the distribution of and market for domestic, commercial and industrial oil burners. This edition is the seventh such survey to be published by this magazine.

The book reviews the developments in the burner industry during the past year, including the production and sales by leading companies, and discusses the competitive situation as it exists among the various fuels. Considerable space is given to a study of the domestic market by states and by type of building heated, accompanied by detailed data on the present distribution of automatic burners. Other subjects covered include the oil price situation, dealer problems, and the market for accessories.

Size 8¾ in. x 11¾ in. Price 75 cents. Can be secured direct from the publisher, 420 Madison Ave., New York. Money must accompany all orders.

John W. Wheller

John W. Wheller, for the past ten years superintendent of the Newark factory of the L. J. Wing Manufacturing Company, died August 18.

Mr. Wheller, who was 35 years of age, had been connected with the company for 17 years, having started as a draftsman.

He is survived by his wife and child, and a brother, Harry S. Wheller, vice-president of L. J. Wing.

Correspondence

A Criticism of the Cities Service Building Hot Water System

EDITOR, HEATING AND VENTILATING:

I have been very much interested in Mr. Space's description of the hot water heating system in the Cities Service Building[1]. In addition to the high static heads involved, the system is interesting because of the complete elimination of any possible trouble with air binding by the very excellent system of venting. Also the use of a high expansion tank with the make-up water admitted directly to the tank is an excellent feature. This arrangement certainly makes for a minimum of corrosion because any water spilled through the overflow is likely to be the new make-up water which contains the most oxygen.

There is, however, one feature of the system that I do not believe will function as planned and that is the method of attempting to reduce the heat input into the south and east sides of the building by means of valves placed in the mains and main risers serving those sides of the building. The trouble with this arrangement is that when the valves are partly closed the distribution of heat to the upper and lower portions of any given zone will not be uniform. In other words, if the engineer succeeds in getting the floors in the middle of the zone to about the right temperature, the upper floors will be too hot and the lower floors will be too cold—a condition which causes both dissatisfaction among the tenants and loss of heat. This condition does not exist when the valves are wide open because the pump is then able to completely overcome the gravity heads generated in the risers and produce uniform distribution.

Perhaps the best way to visualize what happens is to take a numerical illustration. On a clear day with a strong northwest wind, the heat requirements of the south and east sides will probably not be over one-half the requirements of the north and west sides, but in order to give the system the benefit of the doubt, it may be assumed that the requirements of the south and east sides are two-thirds of the requirements of the north and west sides. Other data are as follows:

Temperature of water in flow riser ..... 200°
Temperature of water in north and west return risers.... 170°
Temperature of air in rooms 70°

Since only approximate values are required it will be sufficiently accurate to assume that the heat dissipated by a radiator is proportional to the temperature difference between the radiator and the air in the room.

According to our assumptions the average difference in temperature between the air and the radiators on the north and west sides would be

$$\frac{200 + 170}{2} - 70 = 115° \qquad (1)$$

Assuming that the heat dissipated by a radiator is approximately proportional to the difference in temperature between the radiator and the air, the temperature difference for the radiators on the south and east sides should be:

$$115 \times \frac{2}{3} = 77° \qquad (2)$$

Since the average temperature of the rooms is 70°, the average temperature of the water in these radiators should be:

$$77 + 70 = 147° \qquad (3)$$

To get the temperature in the return risers on the east and south sides let this temperature be t, then since the flow temperature for all sides of the building is 200°:

$$\frac{200 + t}{2} = 147° \qquad (4)$$

And therefore

$$t = 94° \qquad (5)$$

The temperature differentials are therefore: for the north and west sides,

$$200 - 170 = 30° \qquad (6)$$

For the east and south sides

$$200 - 94 = 106° \qquad (7)$$

Having determined the temperature differentials, the next step is to find out whether closing the valves sufficiently to produce the required temperature differentials will cause any short circuits. The first step is to determine how much the flow of water must be decreased, which is determined as follows:

Let H = the heat in B.t.u. dissipated by a radiator per hour,
T = the temperature differential at which the radiator operates, and
Q = the number of pounds of water that flows through the radiator per hour.

Then

$$H = TQ \qquad (8)$$

When the radiators operate at full capacity T equals 30°. This gives

$$H = 30Q \qquad (9)$$

or

$$Q = H/30 \qquad (10)$$

When the radiators operate at two-thirds of full capacity equations (8) and (7) give

$$H^1 = 106\,Q^1 \qquad (11)$$

Where H^1 equals the heat output at two-thirds of full capacity and Q^1 equals the number of pounds of water flowing per hour at two-thirds of full capacity. But

$$H^1 = (2/3)H$$

Whence

$$(2/3)H = 106Q^1 \qquad (12)$$

Or

$$Q^1 = \frac{2H}{3(106)} \qquad (13)$$

Hence dividing equation (10) by equation (13):

$$\frac{Q}{Q^1} = \frac{3(106)}{2(30)} = 5.3 \qquad (14)$$

It should be noted that the quantity of water passing through a pipe or radiator is proportional to the speed of the water, hence V and V^1 may be written in the place of Q and Q^1 in equation (14) which gives

$$\frac{V}{V^1} = 5\,3 \qquad (15)$$

Where V is the speed of the water when the radiators are delivering their maximum heat output, V^1 is the speed of the water when the radiators are delivering two-thirds of their maximum output.

Since the resistance to the flow of water varies approximately as the square of the speed

$$P = V^2R \qquad (16)$$

[1] "A Hot Water Heating System for a 67-Story Building," by J. A. Space, HEATING AND VENTILATING, June, 1932, page 23, and July, 1932, page 31.

Where P is the pressure needed to drive the water through the system when each radiator is delivering its maximum heat output, R is a constant which may be called the resistance of the system when the valves are open.

Similarly, since the pressure generated by the pump is constant

$$P = (V^1)^2 R^1 \qquad (17)$$

Where R^1 is the resistance of the system when the valves are set so that it will deliver two-thirds of its maximum output.

Since P is the same in both equations (being approximately equal to the head generated by the pump) the right hand members of equations (16) and (17) may be set equal to each other giving

$$V^2 R = (V^1)^2 R^1 \qquad (18)$$

or

$$\frac{R^1}{R} = \left(\frac{V}{V^1}\right)^2 \qquad (19)$$

But it has already been shown in equation (14) that

$$\frac{V}{V^1} = 5.3$$

Hence

$$\frac{R^1}{R} = (5.3)^2 = 28.1 \qquad (20)$$

Hence in order to reduce the output of the south and east sides to two-thirds of their maximum output the resistance of the system will have to be made approximately 28 times as great as when the valves are wide open. Or stated in another way, 27/28 of the pressure generated by the pump must be used to overcome the friction of the valves and only 1/28 remains to force the water through the remainder of the system. Since the pump head is 60 ft. the head available for the remainder of the system is

$$60 \times 1/28 = 2.14 \text{ ft.} \qquad (21)$$

The next part of the problem is to find out how great is the gravity head that opposes the pump head in the zone where the radiators are located. Assuming the zone to be 15 stories high and that the height of each floor is about 12 ft., we find that the total height of the piping is about:

$$14 \times 12 = 168 \text{ ft.} \qquad (22)$$

As already worked out, the temperature in the flow riser is 200°, and the temperature in the return riser is 94°. Taking the density of water at 200°, as 60.12 lb. per cu. ft., and the density at 94° as 62.02 lb. per cu. ft., the pressure generated in the zone due to gravity is:

$$\frac{(62.02 - 60.12)168}{62.30} = 5.23 \text{ ft.} \qquad (23)$$

If the friction in the piping below the zone in question and also the gravity head in these pipes which tend to counteract each other are neglected, it can be seen by comparing equations (21) and (23) that the pressure head available from the pump for the radiators at the bottom of the zone is only about 2 ft., while the opposing gravity head is over 5 ft., which would seem to indicate that these radiators will circulate backward, and the same thing is true to a lesser degree of over half the radiation in the south and east half of the zone. However, it should be noted that throughout this analysis it has been tacitly assumed that the system would circulate as planned. The analysis simply shows that this cannot happen, because if it did the water in more than half the radiators would have to flow in a direction opposite to the pressure gradient, which is impossible. What will actually happen is that when the valves are

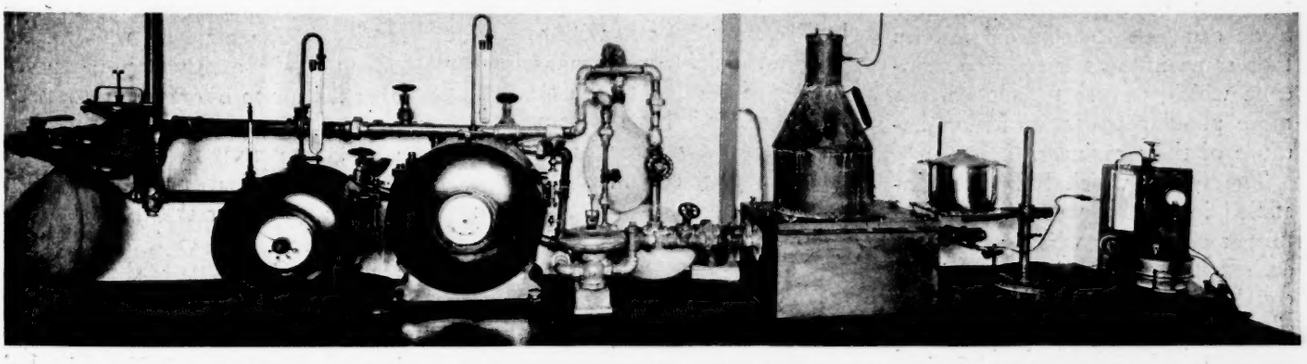
Correspondence

closed sufficiently to appreciably reduce the heat input into half of any zone, the water will not be distributed as planned but the hot water will spread out over the upper floors more or less uniformly and then settle more or less uniformly toward the lower floors, thus making the upper floors too hot and the lower floors too cold.

This trouble can be avoided by using a by-pass and a mixing valve so arranged that for each part of a zone that it is desired to control separately, any fraction of the water from 0% to 100% may be bypassed around the converter. With this arrangement it would be easily possible to reduce the heat input for any zone to less than one-half without any disturbance to circulation. Also the by-pass system is very sensitive. In just a few minutes after the quantity of water bypassed has been increased, the temperature of the radiators makes a sharp drop. The throttling method, on the other hand, is very slow and uncertain even for single radiators.

The simplified layouts in Mr. Space's article indicate that a change to the by-pass system can be rather easily and cheaply made in the case of the Cities Service Building. The writer believes this should be done. It might even be advisable to subdivide each zone into four instead of two parts.

The subject of the effect of gravity heads upon forced circulation hot water heating systems is one that should be given more attention than has been accorded it. The author-



Apparatus used to study the performance of gas burners in the laboratories of the United States Bureau of Standards. The tests, which were conducted by J. H. Iseman, E. R. Weaver and F. A. Smith, were for the purpose of determining the most favorable design of gas burners.

Correspondence

ities on the subject have for many years maintained that in forced circulation systems the gravity heads are negligible, with the result that even the best engineers are usually unaware of the complications that may, under certain conditions, arise if gravity heads are neglected.

In conclusion, the writer would like to state that he considers the heating plant of the Cities Service Building an outstanding engineering achievement and that it points the way toward better heating systems than any that are now available.

ELMER G. SMITH
Dept. of Physics,
Agricultural and Mechanical
College of Texas,
College Station, Texas.

EDITOR, HEATING AND VENTILATING:

I have read with pleasure the comments of Elmer G. Smith on my articles which appeared in the June and July issues of your magazine. At the very outset I wish to take this opportunity to thank Mr. Smith for the interest he has shown and to express the hope that the articles in question elicited an equal amount of interest in the minds of all to whom this matter is of major importance.

I must also plead guilty; but to an indictment which is of lesser degree than would at first be supposed, for my sin has been one of omission rather than of commission. It has been said that brevity is the soul of wit, and to this I might add my own conviction that brevity alone can give to technical articles that same interest sustaining principle which accrues to some of the more popular works of fiction because of their raciness.

The valves which have been installed in both the supply and return mains, and which subdivide each zone into approximately two equal halves, were introduced principally as sectionalizing valves with the thought that for leaks in any of the zone risers or radiator connections—the possibility of which is quite remote because of the materials used in the installation—repairs could be quickly made and involve the necessity of shutting down only one half of each zone. I neglected to mention this fact in the descriptive articles and to this indictment I must of necessity plead guilty.

Of greater importance was my failure to bring out in the descriptive articles the fact that radiation for the entire building has been so proportioned as to take into account the differences in weather which are encountered in this climate for the different exposures, prevailing winds, etc. Installed surface has been very carefully proportioned from data made available by the Weather Bureau, whose records go back over a number of years, with due allowance made for such protection as is afforded by surrounding buildings. For this reason it is felt that regulation between the two halves of any zone would be required only for the very rare occasion when an exceptional cold wave of extended duration would come out of the south. It was also felt that the percentage of variation required at such a time would be very small and that this amount could be obtained by the use of the sectionalizing valves. The regulation would be obtained, however, by manipulation of the valves at the bottom of the zone rather than at the top, as originally stated, because the valves in the return mains are more readily available to the operating force.

I regret very much that my desire to be brief led me to omit from the descriptive articles material which is so essential to a clear understanding of the installation as made, and wish to extend once more my deep appreciation for the interest and technical analysis extended by Mr. Smith. I also wish to extend sincere thanks to your magazine for the space allotted to the articles in question, and hope that you will give these few remarks additional space in your columns.

J. A. SPACE
New York, N. Y.

Information on Government Buildings

EDITOR, HEATING AND VENTILATING:

Whom do I address at Washington to secure information on new Veterans' Hospitals?

Address the Division of Purchases and Supplies, Veterans Administration, Washington, D. C. This division has jurisdiction over Veterans' Hospitals and Soldiers' Homes.

Other departments in which readers may be interested in connection with Federal public buildings are: Construction Department, Quartermaster Generals Office, Munitions Building, Washington, has charge of all buildings for the U. S. Army; Bureau of Supplies and Accounts, U. S. Navy, Washington, has charge of Navy buildings; Superintendent of Construction, Indian Affairs, Department of Interior, Washington, has charge of buildings on Indian Reservations; Supervising Architect, Department of Treasury, has charge of all postoffice buildings, office buildings, federal courthouses, marine hospitals, appraisers' stores, quarantine stations, border stations and custom houses.

Lift Fittings

EDITOR, HEATING AND VENTILATING:

Will you please tell me how it is possible for a lift fitting to operate as shown on page 46 of your August issue, where it says: "Mac sketched the air-loop arrangement for his note book," and also what the necessity is for a lift fitting.

H. M. B.

EDITOR, HEATING AND VENTILATING:

The point in question regarding the lift fitting is one that might reasonably create a question but is explained as follows: The job described in this particular article was one in which a lift fitting was installed. The air loop was merely added to the existing piping to enable the air to by-pass the doorway and, of course, the lift fitting was allowed to remain, this being the most economical procedure. With the air loop over the doorway, the lift fitting ceases to function. Possibly the lift fitting should have been omitted in the sketch, thereby eliminating the possibility of a misunderstanding regarding the functioning of same.

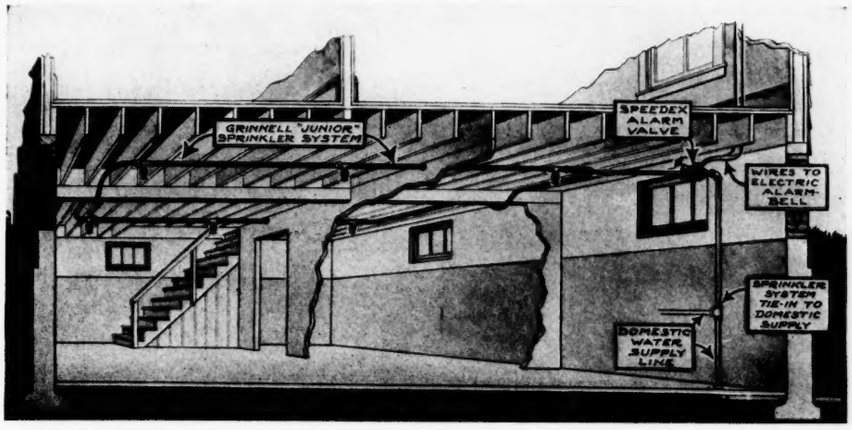
V. D. MILLIKEN

New Equipment

Sprinkler System Protects Homes

Grinnell Co., Inc., Providence, R. I., has announced the Junior automatic sprinkler system for fire protection in residential basements, small stores, school houses, and other occupancies of limited area.

These sprinkler systems are connected to the domestic water supply and are piped with flexible copper tubing installed in a loop. This provides for more even distribution of water and pressure no matter where the operating sprinkler may be.

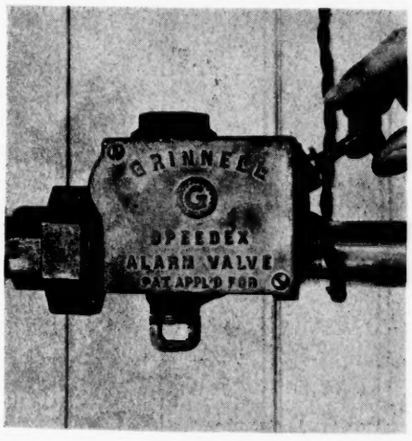

GRINNELL "JUNIOR" SPRINKLER SYSTEM
SPEEDEX ALARM VALVE
WIRES TO ELECTRIC ALARM BELL
SPRINKLER SYSTEM TIE-IN TO DOMESTIC SUPPLY
DOMESTIC WATER SUPPLY LINE

Grinnell Junior sprinkler system

Speedex sprinkler heads used with this system are attached in the pendent position, which facilitates erection of the tubing by permitting it to be suspended directly from the ceiling by means of clip hangers. These heads are held tightly shut by a quartzoid bulb which will open only when subjected to a temperature of 135°. When the bulb opens water issues from the tubing and is broken up by the heads into a pressure-driven rain.

As soon as water flows through the tubing to an open head, an alarm is automatically sounded by the Speedex alarm valve. This valve, a combination alarm, check, and gate valve, is installed between the domestic supply connection and the loop of the sprinkler tubing. Ordinarily the check in the valve is closed in the water way, but when a head opens the rush of water moves it out of the way and it acts to establish an electric current which rings the bell and announces the fire.

Tubing is supplied in 12 ft. lengths. On one end of each tube a Speedex sprinkler fitting is attached. This fitting carries a tubing union on one end and also a threaded outlet to receive the sprinkler head. For installation in locations where high temperatures ordinarily occur, heads rated for 175° may be had.

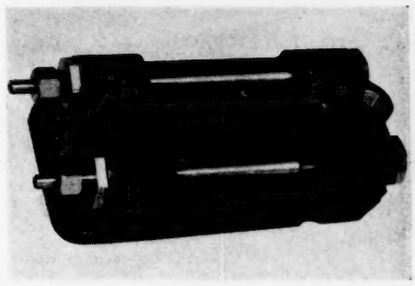


Alarm valve

Oil Valve Compensates for Viscosity Changes

Ray Burner Corp., 401 Bernal Ave., San Francisco, is equipping its automatic burners with the Ray viscosity valve system. This appliance regulates the flow of oil to the burner to maintain a constant supply with variations in oil viscosity.

Mechanically, the device consists of two taper valves and a relief valve incorporated into a single cast iron body, mounted above a small cast iron oil reservoir, and connected to a constant discharge pump. One tapered valve is in the line of flow from the reservoir to the burner, and the other is in the return line from the burner. No other metering device is required.



Cutaway view of viscosity valve

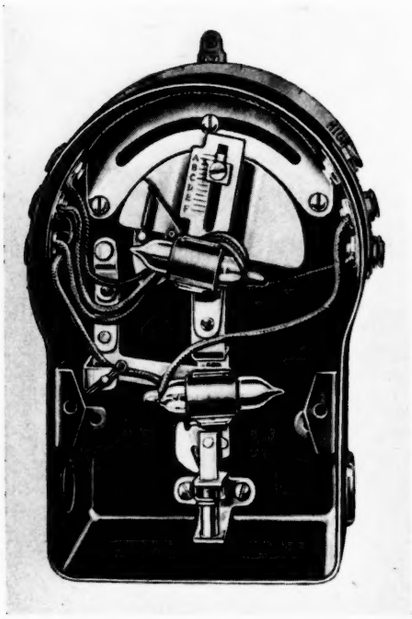
When increased viscosity of the oil restricts the passage of the oil through the first viscosity valve and to the burner, the second valve sets up an equal restriction to flow of oil away from the burner and back to the tank. Since the pump is practically a constant delivery medium, the result is an increase in pressure under which the fuel is delivered. This increase in pressure insures a constant delivery rate to the burner regardless of variations in the viscosity of the oil. The relief valve confines the pressure within safe limits. Both valves are adjustable, and it is possible to regulate what percentage of the oil delivered by the pump shall flow to the burner, and what percentage shall be returned to the tank.

This system of fuel supply makes possible the burning of oils as heavy as A.P.I. Specifications No. 5 and No. 6, which flow sluggishly when cold and freely when warm.

Control Maintains Fire During "Off" Periods of Automatic Stokers

Minneapolis-Honeywell Regulator Co., Minneapolis, Minn., is manufacturing the Type L Stoker-switch which in conjunction with a room thermostat controls the operation of domestic and small industrial stokers. Its function is to prevent the fire from going out by starting the stoker for brief feeding operations as the temperature in

New Equipment



Stokerswitch with "outfire cut-off"

the stack falls during long "off" periods of the room thermostat.

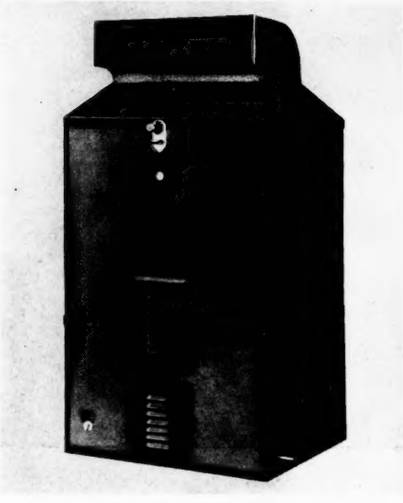
In operation the pointer on the outside of the case is set on the dial for the lowest stack temperature which will operate the stoker just long enough to pick up the fire when it gets too low. When the stack temperature has increased the necessary amount, the mercury switch opens to break the circuit and stop the stoker. The number of degrees between the points at which the Stokerswitch will start and stop is adjustable by means of the graduated plate and set screw adjustment inside the case, which provides for a differential of from 25° to 200°.

These switches are available in models for use with both high and low voltage controls. They may also be had with the "outfire cut-off" as an additional feature. In this case a second mercury tube switch with suitable linkage is used to open the motor circuit to prevent operation of the stoker by either the thermostat or the Stokerswitch in the event that the fire becomes extinguished, as indicated by a stack temperature below the minimum temperature. When the fire has been rekindled, the control is reset manually by means of a button located at the bottom of the case.

Air Conditioning Gas Furnace

A gas-fired air conditioning warm-air furnace, the Niagara 110, has been announced by the Forest City Foundries Co., 2500 West 27 St., Cleveland.

Units are built in two compartments, the upper containing the heat generating equipment, radiating sections, controls, and humidifier, while the blower and filter equipment is located in the lower section. Air enters the lower compartment through filters and is delivered by two variable speed blowers to the upper compartment where it is warmed by contact with the walls of the radiating sections.



Niagara 110 air conditioning furnace

Units are built in five sizes, ranging in output from 72,000 B.t.u. to 176,000 B.t.u. per hr. when gravity circulation is used, and from 76,500 B.t.u. to 187,000 B.t.u. per hr. under forced circulation.

Fitting Protects Threaded Ends of Pipe

Sarco Company, Inc., 183 Madison Ave., New York, is marketing the Pipe Saver, a flanged bushing which fits snugly into the threaded end of a pipe to protect this weak point against the action of corrosive elements in steam, water, oil, or chemicals.

Bushings have tapered ends and are shaped with a bulge at the middle which forces the tapered end against the pipe when the Pipe Saver is driven into place. The flange is seated tightly against the end of the pipe and is threaded into the fitting with the pipe. The bushings are made in sizes ranging from ½ in. to 8 in. for standard pipe, and in sizes from 1¼ in. to 6 in. for extra heavy pipe. They are available either in brass or heavy chromium plate.

Foxboro Pneumatic Switch

A compact pneumatic switch has been designed by The Foxboro Co., Foxboro, Mass., for use with its standard temperature and pressure controllers.

Of the standard single pole mercury type, the switch is mounted in a small cast iron housing with an outlet tapped for either conduit or BX connector. A small diaphragm motor, mounted directly on the housing, operates the switch.

Radiant Panel Heating Elements Contained in Screen

A screen wired for low temperature electric panel heating has been developed in the laboratories of the Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

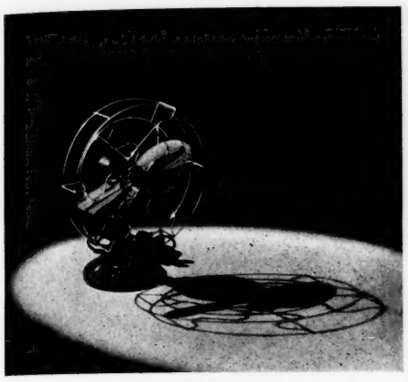
These screens are said to maintain an effective temperature within the comfort range in mild weather without the use of other heating provision, and will also maintain comfortable conditions in extremely cold weather when the outside walls are insufficiently heated by the existing heating apparatus. By reducing the proportion of heat transferred by radiation from the occupants to the walls, it is possible to maintain com-

Westinghouse radiant panel screen

fortable conditions in the room with a lower general temperature. The screen shown, said to be larger than would ordinarily be necessary, consumes 750 watts.

Barcol Fan

Pleasing appearance, combined with quiet and efficient operation, makes the Barcol Eight fan, manu-



Barcol Eight fan

factured by the Barber-Colman Co., Rockford, Ill., a desirable appliance for home and office.

Case and standard are finished in stainproof phenolic resin, and the guard and blades are cadmium-plated. Eight inches is the distance between the tips of the two-bladed fan. Four blade models, 6 in. in diameter are also available.

Control-Aire Portable Humidifier

Hexcel Radiator Co., Milwaukee, Wis., is manufacturing the Control-Aire portable humidifier for domestic, office, and similar uses.

An electric heating element brings water in the reservoir to the required temperature for evaporation. Air is drawn in at the back of the unit, and is passed through a vapor chamber from which it is expelled into the room. A switch makes possible two rates of operation. At the higher rate, approximately 1 gal. of water is evaporated in 2½ hr., and at the lower switch setting 1 gal. is evaporated in approximately 3½ hr. An automatic cut-off stops the dehumidifier when the water supply is exhausted, and the unit can be turned on and off at will by means of the switch.

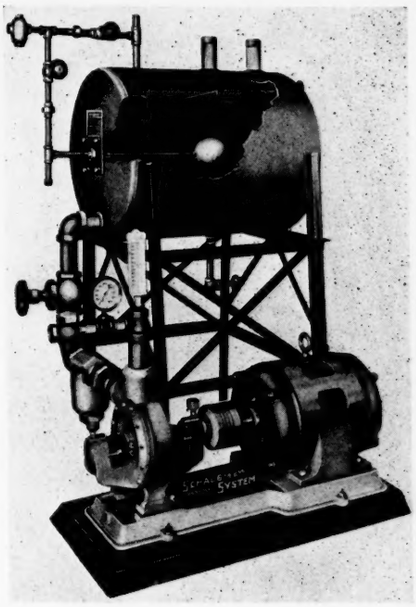
Units are designed for 110-volt circuits, and may be had for either direct or alternating current.

Schaub Boiler Return System

Fred H. Schaub Engineering Co., 325 West Huron St., Chicago, is manufacturing the Schaub boiler return system, a unit which automatically maintains a uniform water level in the boiler, supplies make-up feed water, and returns high and low pressure condensation.

Water is admitted to the boiler in exactly the proportion that steam is used, and at the same time the temperature of the water supplied is maintained constant. This results in a uniform steam space in the boiler, preventing wide fluctuation of steam pressures, and saving both fuel and labor in the operation of the plant.

A float valve in the receiver con-



Automatic boiler return system

trols the amount of water added to the condensate. This valve is cone shaped and made of hard valve steel. The seat is of the same material, but softer, so that wear is on the seat. The boiler level control consists of a float operated Mercoid switch which is located on the boiler at the water level.

Schaub return systems may be had in sizes for boilers ranging in capacity from 1 hp. to 300 hp.

Ice Room Cooler Washes and Purifies Air

The Corozone Co., Hanna Building, Cleveland, is marketing the DeLuxe Aquazone room cooler and air conditioner, using ice as the refrigerant.

Air is brought into the device at the rate of 78,000 cu. ft. per hour. Chilled air from the ice chamber is added, and the air is then forced through the refrigerant wheels. These refrigerant wheels, two in number, consist of wire mesh screens 16 in. in diameter to which are fastened eight curved water dippers which pick up and discharge water over the wheels. The air is both cooled and washed by this device. It is then compressed and passed into finned water tubes where it is further chilled and dehumidified before being discharged back into the room.

Deodorization and revitalization are accomplished by the introduction of ozone from the Corozone, a device which generates this material by means of a corona discharge.

Relative Humidity Calculator for High Temperature Work

J. B. Maas & Co., Fox Creek Sta., Detroit, Mich., is marketing the Humidiflash, a device combining a relative humidity calculator with a pocket flashlight.

A stationary scale at the top and bottom of the flash light barrel is marked for dry bulb temperatures ranging between 100° F. and 200°. On the revolving outer sleeve are arrows marked for wet bulb depressions or dry and wet bulb temperature differences. When the

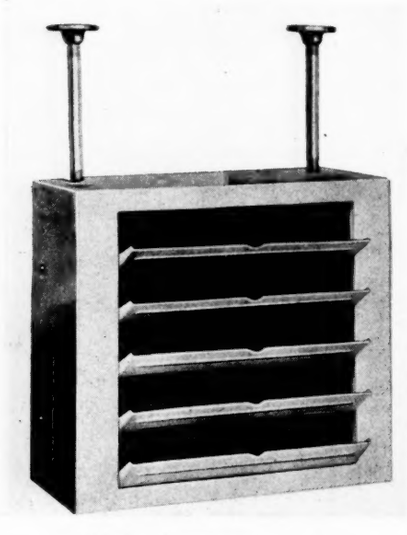
Humidiflash

psychrometer or hygrometer readings have been taken, the arrow corresponding to the wet bulb depression is set on the dry bulb temperature on the stationary scale, and the percent relative humidity is read through the aperture for the selected arrow. The manufacturer states it is of value in dry kiln and other process work.

Combination Cooling and Heating Unit

McCord Radiator & Mfg. Co., Detroit, has developed a convector that performs the cooling, dehumidifying, circulating, and heating functions of air conditioning. It is recommended by the manufacturer as finding application in business and professional offices, stores, restaurants, banks, and similar uses.

Units consist of a rigid chassis, supporting a cellular copper radiator core in an insulated shell behind which an electrically-operated air impeller is mounted on rubber. The convector is connected to a suitably located main cooling system in which water is cooled either by ice or mechanical refrigeration. Adjustable louvers in the discharge of the convector may be set to give the cooled air stream any desired elevation. Excessive humidity is condensed out of the air by contact with the cold convector coil, from which it immediately drains. During mildly warm weather, the desired cooling effect may be obtained without ice or refrigeration by connecting the cooling tank to the regular cold water supply. The complete unit is so constructed that its cooling capacity is under full control.

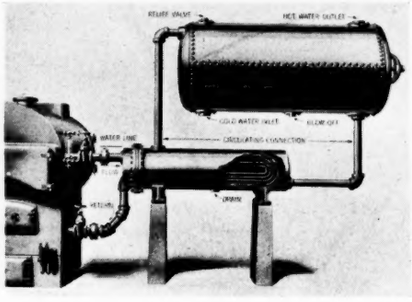


McCord convector unit

For winter operation, the convector is connected to a suitable hot water supply. The unit is suspended from the ceiling or built into the walls of new structures.

Patterson-Kelley Indirect Water Heater

A hot water service heater, intended to be used as an indirect circulating heater, has been announced by The Patterson-Kelley Co., 101 Park Ave., New York, N. Y. It is called the Type D Lo-Flo heater, and its design is said to be such that almost any condition of installation can be met.



Patterson heater connected to boiler and tank

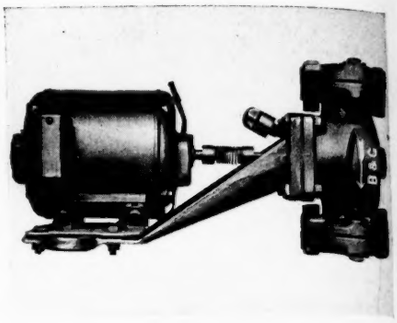
A steel jacket encloses the heater, and all joints are electrically welded. The tube sheet is of heavy forged steel, and both sides of the sheet are machined to insure tight gasketed joints. The flow chamber is of cast iron, and the heating surface is made up of seamless drawn copper tubing shaped to form U-bends, with both ends rolled into the tube sheet.

Heater is installed below the water line as close to the boiler as is possible and connected to it by flow and return pipes. The boiler water preferably circulates through the shell, and the tubes are connected to the storage tank.

These heaters are said to be particularly suited to installations operating under high water pressure and are adapted for use in connection with existing storage tanks. Seventeen sizes of heater are made, ranging in capacity from 40 gal. per hr. to 2400 gal. per hr., based on a temperature rise of 100° with boiler water at 180°.

Centrifugal Circulator for Hot Water Systems

Bell & Gossett Co., 3000 Wallace St., Chicago, has announced the B & G Booster, an electrically-



Bell & Gossett Booster

driven circulator with positive centrifugal pump action for hot water heating or domestic hot water service.

A bronze centrifugal rotor is enclosed in a flange type cast iron pump casing. This casing is connected to the base by four bolts, providing a simple means of setting the pipe connections for either horizontal or vertical installation. A stainless steel shaft, operating in graphite impregnated bearings, drives the rotor. The motor is mounted horizontally with rubber insulators separating it from the base.

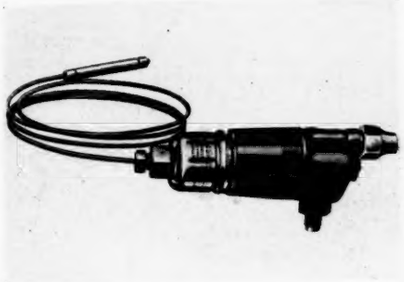
Circulators are supplied in two sizes, discharging 30 g.p.m. and 50 g.p.m., and having 1/8 hp. and 1/6 hp. motors, respectively.

Thermostatic Expansion Valve

Detroit Lubricator Co., Detroit, is manufacturing the No. 673 Genuine Detroit thermostatic expansion valve. This device is to succeed the No. 671 American thermostatic expansion valve widely applied in mechanical refrigerating systems using methyl chloride, sulphur dioxide, and F-12.

A thermostatic bulb is clamped to the suction line as near as possible to the point where it connects to the cooling coil or to the end of the cooling coil proper. This bulb is charged with a thermostatic liquid which

creates a vapor pressure in what is called the thermostatic power element, within the valve body. This power element works through a push rod against the regular bellows in the valve. Thus the valve operates on the difference between the pressure (or temperature) of the refrigerant in the low side and the temperature at the bulb. When the



Detroit thermostatic expansion valve

bulb is properly clamped to the suction line, the valve operates to maintain a constant degree of superheat at that point. Valves are adjusted at the factory so that the coil is kept full of refrigerant without any possibility of frost back.

Valves are furnished standard with 0.078 in. orifice diameter, but on special order can be had with 0.125 in. or 0.156 in. orifice. The 0.078 in. orifice has a capacity of two tons of methyl chloride, and one ton with sulphur dioxide or F-12 refrigeration. The 0.125 in. orifice increases these capacities by 50%, and the largest orifice has a capacity of four tons of methyl chloride and two tons of sulphur dioxide or F-12 refrigeration.

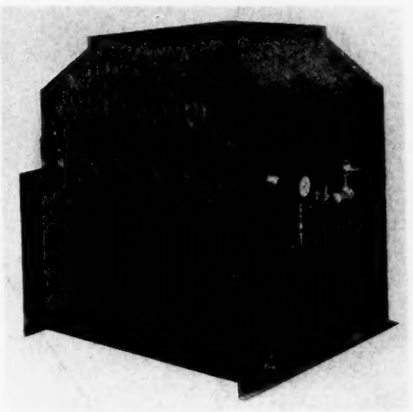
Worthington GA Water Meters

Worthington Pump and Machinery Corp., Harrison, N. J., has added the Model GA water meter to its line. It is said to be accurate for extremely low flows as well as for greater volumes.

The device is of the positive displacement disc type. All register gears are fitted with rubber bushed bearings, and the train assembly comprises four cast bronze gears and four pinions, rigidly fastened to Monel metal spindles. The measuring chamber is made in two sections, held together by a snap joint without screws, permitting the case to separate without distortion in the event of freezing. Additional protection is provided by a cast iron frost bottom in the smaller sizes. This bottom breaks when water in the meter freezes, relieving strain on the interior parts. Stuffing boxes are cast on the main case, eliminating gaskets and lock nuts, and preventing a possibility of leakage from this source.

Climator III Air Conditioning Unit

L. J. Mueller Furnace Co., Milwaukee, Wis., has announced the Climator III, a unit combining fan, air washer, and filters for supplying air conditioning with any type of warm-air furnace either in new or



Mueller air conditioning unit

existing installations. The unit is of such dimensions that it can be taken through a 2½ ft. door without disassembling.

A large diameter fan provides sufficient air flow to permit the use of efficient filters and washer, and the connection of a system of recirculating ducts.

Electric Timing Device Controls Day and Night Operation

An electric control system, called the Selective Syncrostat, for use in connection with its automatic stoker has been announced by The Iron Fireman Mfg. Co., 3170 West 106th St., Cleveland. This control unit, operated by a Telechron electric clock, automatically regulates day and night temperatures in the home, and in addition maintains fire in the heating plant regardless of weather conditions.

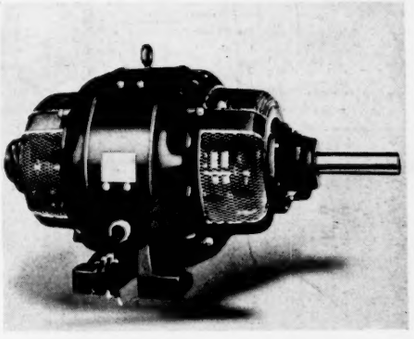
Two instruments are included in this control. A duplex thermostat containing two separate elements, one for day temperature and one for night temperature, is set by means of a knob dial. This thermostat is connected directly to the time switch which regulates the temperature of the home to the desired point for day and night operation by transferring the control from one to the other of the thermostatic elements at stated intervals. The transfer switch is actuated by two hands which revolve with the 24-hour time dial and transfer control from day to night temperature, and vice versa, at any hour depending on their setting.

This timing device also provides for refueling to prevent the fire from going out during periods of mild weather. The length and frequency of the refueling periods are regulated by settings made on two dials on the time switch. In this way, just sufficient fuel can be fed to the fire to keep it alive and adjustment can be made for differences in weather conditions and coal characteristics.

Ideal Self-Syn Motor

An improved synchronous motor for industrial and general purposes has been developed by The Ideal Electric & Mfg. Co., Mansfield, Ohio. It is called the Self-Syn motor.

It is a compact synchronous motor, with a simplified control. The exciter windings and direct current commutator are integral with



Self-synchronizing motor

the unit, and no external excitation is necessary. Since the motor is self-synchronizing, no synchronizing relay or field switch is required, and the machine is started with a simple hand operated compensator in the same manner as an induction motor. The manufacturer also states that the motor will re-synchronize itself after having been pulled out of step by a line voltage dip or a momentary overload.

Motors are supplied in all sizes from 5 hp. to 100 hp. Power factors of unity or 80% leading are standard.

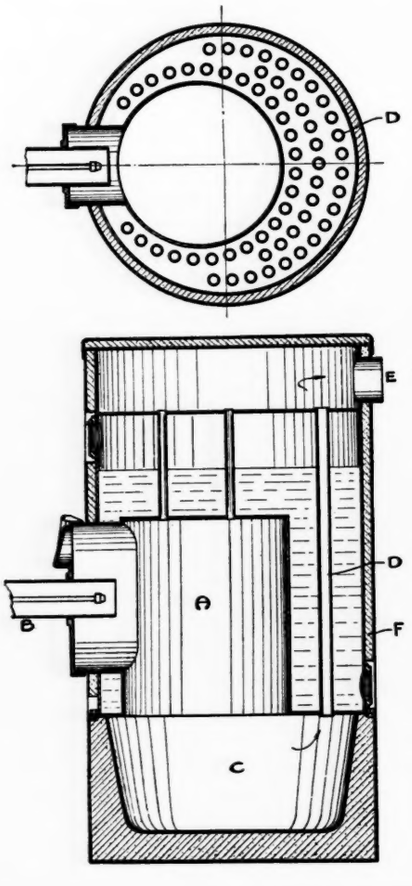
Bass Oil Burning Steel Boiler

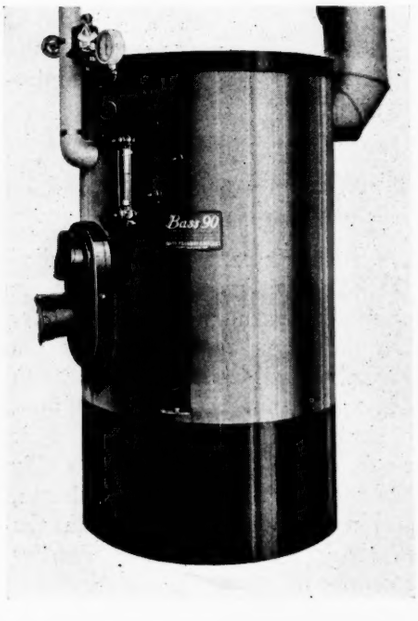
Bass Foundry & Machine Co., Fort Wayne, Ind., has announced the Bass 90 oil-fired steel heating boiler for steam and hot water heating systems. It may be used with any oil burner of the gun type.

Burner nozzle is shown at B, and combustion takes place in the upper chamber represented by A. The hot gases pass downward from the combustion chamber into the reflector base C, which is a bowl-shaped compartment lined with insulating cement and surfaced with refractory cement. From here they pass upward through the copper tubes D, and out the smoke outlet E.

Steel plates 5/16 in. thick are used in the boiler proper, which is of welded construction. The sides, top, and bottom are insulated with rockwool F, covered with a steel jacket. Boilers are tapped for the installation of an indirect hot water heater.

Standard finish is blue enamel with black trimmings, but special color combinations are obtainable. The boiler is available in sizes to take care of steam radiation loads up to 2500 sq. ft., and for hot water loads up to 4000 sq. ft. of radiation.





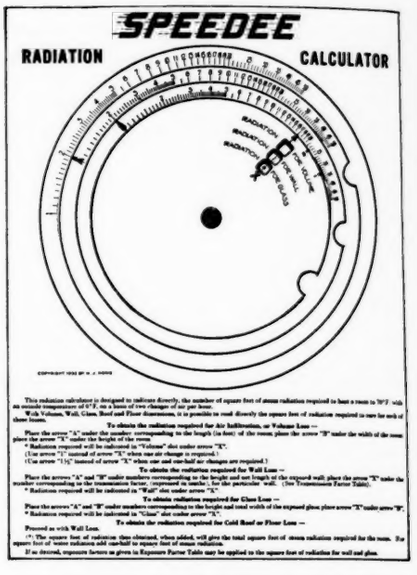
(Left) Plan and elevation sections of Bass boiler. (Right) Bass oil-fired boiler

Rapid Reagent Developed for Analyzers

$SeizO_2$ is the name of an extremely rapid oxygen absorbing material developed by The Hays Corp., Michigan City, Ind., for use in gas analyzers. It is a stable alkaline solution, unaffected by light and capable of absorbing 47 times its volume of oxygen. The manufacturer states that the chemical will remove oxygen from about 500 samples of flue gas, and loses its efficiency abruptly when it has reached its capacity.

Device Computes Radiation Requirements

Norman J. Howe, 75 Maple St., West Roxbury, Boston, Mass., is manufacturing and distributing the Speedee radiation calculator. This instrument indicates directly the



Speedee radiation calculator

number of square feet of steam radiation required to heat a room to 70° with an outside temperature of 0°, on a basis of two changes of air per hour. With volume, wall, glass, roof, and floor dimensions known, it is possible to read directly the square feet of radiation required to care for each of these losses.

Three cardboard discs are mounted concentrically on a pivot on a heavy cardboard back. Dimensional settings are made by rotating the discs so that the arrows are brought under the stated numbers on the scales, and the required radiation is read in the proper slot in the top disc. Full directions are printed on the disc. On the reverse side of the calculator are printed a table of transmission factors for various structural materials and types of construction, a table of exposure factors, and a table of factors for radiation required for rooms heated to other than 70°. There is also provision for calculating the radiation directly based on one, and one and one-half, changes of air hourly.

Vacuum Refrigerating Unit for Air Conditioning

Ross Heater & Mfg. Co., Inc., 1407 West Ave., Buffalo, N. Y., is manufacturing refrigerating units operating on the steam ejector principle. They are applicable to any work in which water at moderately low temperatures is required.

Steam at any available pressure may be used. Water to be cooled is sprayed into the evaporator or flash chamber in which a high degree of vacuum is maintained. A small amount of this water is immediately flashed into steam, absorbing its heat of vaporization from the remaining water. The chilled water is withdrawn from the bottom of the evaporator and circulated to the point at which it is to be used.

Vacuum is maintained in the evaporator by a series of air ejectors and condensers. The primary ejector compresses the steam which has been flashed in the chamber plus the air which has entered the system with the water to a slightly higher absolute pressure. In the primary condenser, the steam from the evaporator and the propelling steam are condensed, and the air further compressed reaching atmospheric pressure in the secondary ejectors. An "inter-and-after" condenser condenses the propelling steam of the secondary ejectors.

Condensing water may be taken from city mains or circulated in a cooling tower. The primary condenser and the inter-and-after condenser are generally of the surface type, but where conditions make it advisable, barometric condensers may be used.

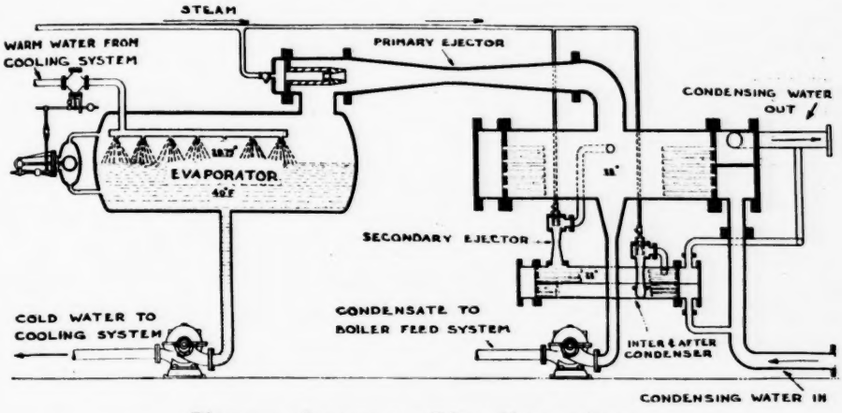
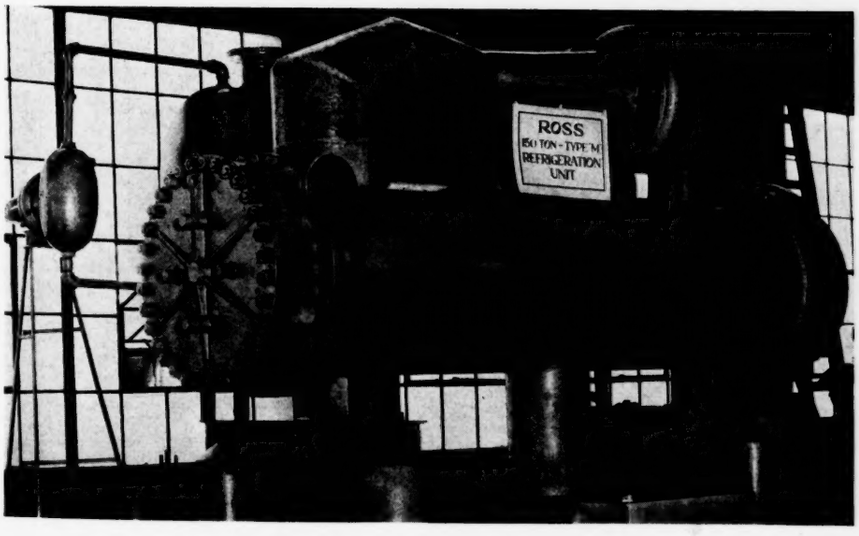


Diagram of vacuum refrigerating system

These units will produce chilled water at temperatures as low as 32° F. For temperatures below 40° a very high degree of vacuum must be maintained in the evaporator. Moderate vacuums will produce chilled water between 40° and 60°.



Ross 150-ton vacuum refrigerating unit

Arrangement of the various parts of the unit may be modified to comply with space limitations in the building in which it is installed.

Radiator Heats by Both Radiation and Convection

Shaw-Perkins Manufacturing Co., Oliver Building, Pittsburgh, Pa., is manufacturing the Shaw radiator, consisting of a copper tube convector enclosed within a steel extended surface which is fitted with grilles. The steel surfaces are in contact with the copper tubing, and heat is given off by convection to the air which circulates through and over the radiator as well as by radiation from the exterior surfaces. The radiator, therefore, forms its own cabinet, grille, and radiator, all in an integral unit.

Units are furnished with either front or top air outlet grille, and several different applications are possible. Installed fully exposed upon legs or wall brackets, it provides its own radiant cabinet and grille. When installed recessed in the wall with the front exposed for radiant heat and appearance, it serves as radiator, radiant front panel, enclosure, and grille. It may also be recessed and concealed with ordinary recessed enclosures.

Heating medium, water or steam, is confined within non-corrosive heavy gauge copper tube, and the radiator can be used with high pressure steam or high pressure forced circulation hot water, as well as with low pressures. The exterior surfaces form a protective shield against the high temperatures of high pressure work.

Other advantages mentioned by the manufacturer are its lightness, compactness, ease of cleaning, and the wide range of decorative treatments made possible by its restrained modern lines and variety of finish.

Units are made in four heights, from 14 in. to 23 in., in both the top air outlet and the front air outlet models, and are 3 in. wide. Capacities range from 3.4 sq. ft. to 60.4 sq. ft. per radiator.

Business Statistics

(Dotted line in charts, 1931; solid line, 1932)

Building permits issued during July in 215 cities were valued at $26.8 million, as compared with $32.2 million in June, a decrease of 16.2%, according to *Bradstreet's Weekly*. The decline was more than seasonal, but this is tempered by the fact the month had one less business day than July in other years.

Bradstreet's notes one unusual feature in the July figures, the fact that contracts for that month increased over June while permits continued to decline. Apparently this indicates that a larger number of proposed projects are reaching the construction stage than previously.

Poor's Index of General Business rose to 69.9 in July, according to the preliminary estimate in Poor's "Business and Markets" bulletin. This is the fourth successive monthly rise shown by this index, which started its upturn in April. The index, as explained last month, differs from some other indicators of general business activity in that it omits freight car loadings, an item whose significance Poor's believes has been largely impaired in recent years by the inroads of truck competition on the freight business of the railroads, although car loadings are still carefully followed.

The rise in July, according to Poor's, "is due mainly to the fact that July is normally a low month. The declines in production in the steel and automobile industries were not so great as seasonally expected. Mail order house sales also declined less than their seasonal index. Electric power moved slightly upward, contrary to its seasonal pull. Crude oil production advanced somewhat, in accordance with its seasonal trend."

Employment in the steamfittings manufacturing plants during July stood at 32.5% of the 1926 average, as compared with 33.7% for June, according to the Department of Labor. Payrolls in the same plants were at 18.5% of the 1926 average, as compared with 33.7% for June. Both employment and payrolls in such plants dropped less from June to July than in the average of all manufacturing plants.

The heat loss of a typical residence, according to C. George Segeler, American Gas Association, is proportioned as follows: roof or ceiling, 14.8%; walls, 29.6%; glass and doors, 26%; floor, 8.8%, and infiltration, 20.8%. The percentages were calculated from data on 400 dwellings.

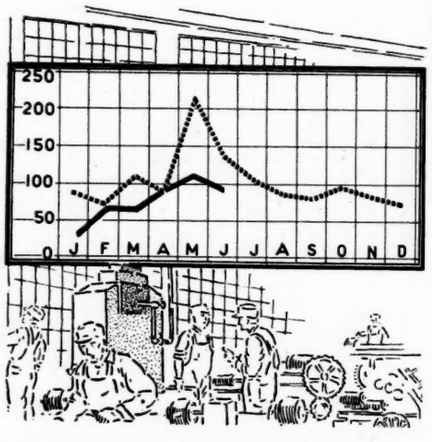
Building permits for the first six months of 1932, as compared with the same period last year, show that new residential building has declined 76%, non-residential building dropped 73.5%, and total construction, including alterations and repairs, fell off 60.6%.

New building contracts awarded during July amounted to $128.8 million, as compared with $113 million for June, an increase of 14%. As compared with July of last year, however, the July, 1932, figures show a decrease of 55%.

The valuation of contracts for July consisted of $19.7 million for residential building, $48.9 million for non-residential building, and $60 million for public works and utilities.

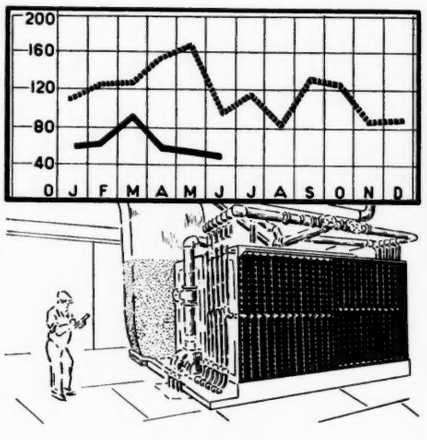
Climbing consistently since April, construction contracts in 16 southern states reached $213 million for the first seven months of the year with nearly $46 million in new contracts added during July, the *Manufacturers Record* says in its August issue.

The $46 million figure exceeds the April figure by 76%, the May total by more than 30% and the June total by more than 5%, the *Record* says.



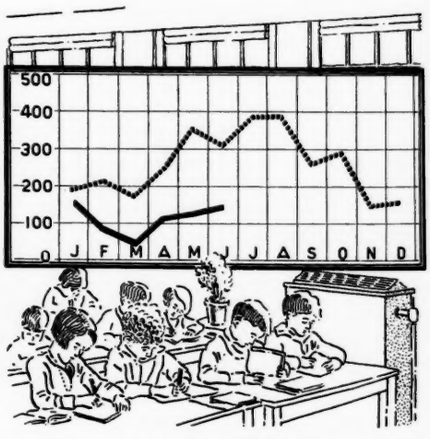
Air Conditioning Units and Washers
Orders in Dollars

June, 1932, 93,215; June, 1931, 138,858; decrease from last year; 32.9%. Total, six months, 1932, 461,245; total, six months, 1931, 716,384; decrease from last year, 35.6%.



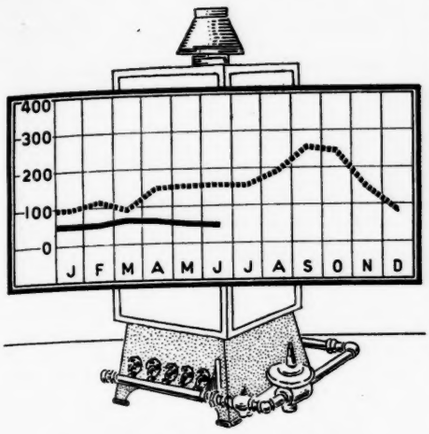
Indirect Heating Surface
Orders in Dollars

June, 1932, 50,071; June, 1931, 96,538; decrease from last year, 48%. Total, six months, 1932, 374,094; total, six months, 1931, 778,404; decrease from last year, 46%.

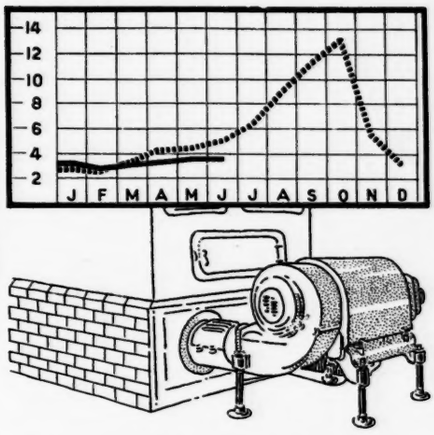


Unit Ventilators
Orders in Dollars

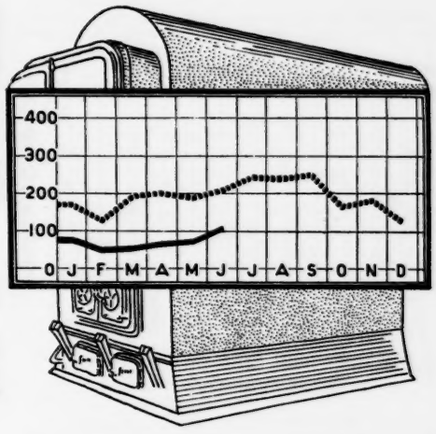
June, 1932, 140,968; June, 1931, 312,387; decrease from last year, 54.7%. Total, six months, 1932, 665,637; total, six months, 1931, 1,502,604; decrease from last year, 55.6%.



Shipments of Gas-Fired Boilers
In Thousands of B.t.u. Capacity
June, 1932, 48,821; June, 1931, 156,769; decrease from last year, 68.5%. Total, six months, 1932, 344,928; total, six months, 1931, 770,401; decrease from last year, 55.2%.



Domestic and Commercial Oil Burners
Shipments in Number of Units
June, 1932, 3702; June, 1931, 4974; decrease from last year, 25.6%. Total, six months, 1932, 19,967; total, six months, 1931, 22,228; decrease from last year, 10.2%.



New Orders for Steel Heating Boilers
In Square Feet of Heating Surface
June, 1932, 102,788; June, 1931, 205,441; decrease from last year, 50%. Total, six months, 1932, 418,996; total, six months, 1931, 1,084,551; decrease from last year, 61.4%.

By including numerous small projects for which contracts were awarded but which were not reported separately, the seven-month total is conservatively estimated at $280 million.

In Washington a contract involving over $1 million was let for steam lines and tunnels forming part of the giant central heating system for government buildings.

May figures for the manufactured gas industry, as compiled by Paul Ryan, statistician for the American Gas Association, show an increase of 11.4% in the number of house-heating customers, as compared with May of 1931. The data cover 206 companies.

The long descent in construction costs prevailing since 1924 has apparently been checked, according to the construction cost index prepared by *Engineering News-Record*. The index, based on 1913 costs equal 100, stood at 152.20 for June, the lowest in 16 years. In July the figure rose to 153.36, and for August climbed to 156.80. The last increase was due to a considerable rise in cement prices, which have been boosted 15 cents to 50 cents per barrel in many sections of the country. The cost index is made up from the cost of common labor and the price of three basic commodities, cement, steel and lumber.

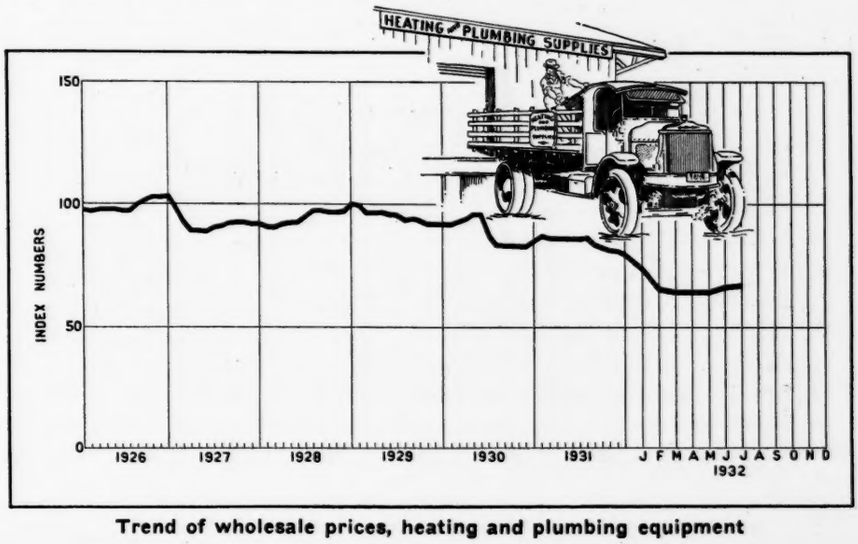
Indicated expenditures for building operations in June were $49 million in 354 cities, according to the Bureau of Labor Statistics. This is 26.3% less than for May. New residential permits increased 0.3%, while non-residential permits declined 38.3%. There was an increase of 2.7% in the amount spent for additions, alterations and repairs.

As compared with May, the June figures for new buildings show some interesting changes. Single family dwellings dropped 18.5%, two-family houses dropped 30.9%, while apartments houses and flats increased 386.3%. However, as compared with June, 1931, single-family houses dropped 71.8%, two-family dwellings declined 76.1% and multi-family buildings dropped 85.3%.

Employment in building construction decreased 4.3% in June, as compared with May, while earnings decreased 6.2%, according to the Department of Labor.

Material prices for heating and plumbing equipment increased slightly during July, according to the index of wholesale prices prepared by the Department of Labor. The index stands at 67.1 for July, 100 being the 1926 average. The July figure shows an increase of 0.4 over the June figure, and an increase of 2.7 over May. The continued increase gives a slightly optimistic tone to the price situation, inasmuch as it may reflect a somewhat quickened demand.

Building materials as a whole, however, dropped off still further from 70.8 in June to 69.7 in July. As the index shows, heating and plumbing materials have dropped further in price than other building materials since 1926. This is particularly true of structural steel, the July price of which is at 81.7% of the 1926 figure.



Trend of wholesale prices, heating and plumbing equipment

Skinny Business

By Ab El Pilauf

ACHMED looked around his bazaar and mused verbally. "Yea, I shall have to grab me a new graft, now that the followers of Mahomet have gone on a buyers' strike against wind pushers and braziers. Methinks I shall found a new industry and apply quantity production methods to the manufacture of cat skins.

"Allah, in his wisdom, hath given a cat nine lives. The hazards of spoilage, therefore, are ninefold removed. In order that the cats may increase and multiply, I shall make a dicker with Muley-Hamed for the rats that infest his granary.

"Now, as these rats increase and multiply fourfold as fast as cats, behold, I shall always maintain an ever-increasing supply of raw material for my ever-increasing herd of cats."

ACHMED AND HIS CATS

And calculating the profits that would be his from his flock of skins at ten shekels per skin he became dazed at the prospect of his riches, and gloated.

Soon his competitors were astounded to see him open a cat ranch, and to all their commiseration Achmed spake, "When thy business goeth on the bum, seek thou a strange business." And they went their way, muttering. It was even as Achmed spake. When the wind pusher and brazier business of his competitors went to the dogs, Achmed went after the cats. And in all the land of the prophet there was not the equal of his cat crop. Bartering his stock and tools for still more cats, he finally sold his bazaar and bought an abandoned silver fox farm that had perished because of the immutable fixity of biological and financial laws.

His competitors, forsooth, when they saw the collapse of the market for wind pushers and braziers, got themselves busy, and harnessing the harmattan, which was the wild desert wind, and combining it with the cool water of the wells of the oasis, a new industry was born that cooled the tents of the faithful and caused them to rejoice that Allah had given them this new comfort, and this new wealth that was now their lot.

But Achmed stuck to his cats, saying, "When the good old days come back, I shall reopen my shop." But the good old days were gone forever. Years later Achmed staggered back to the market place, and was astounded at the things he saw. Copiously, his competitors had prospered and there was no more of the old-time grief.

"How farest thou, Ben-Ali?" said Achmed to the first one he met.

"Oh, swell!" replied Ben-Ali. "And how dost thou get on with thy cat farm?"

"Twas a noble idea," said Achmed, sadly. "That is what ruined it, yea, even a nobler idea. It seemed a wanton waste to kill those cats for their pelts, so I crossed them with snakes who shed their skins twice a year, whereupon the market folded up and so did I."

And Achmed was glad to get a job from Ben-Ali as a helper at sixty shekels per hour.

With the Manufacturers

New Castle Plant of National Radiator Sold

Properties in New Castle, Pa., of the National Radiator Corp., Johnstown, Pa., were purchased from the receivers on August 8 by the reorganization committee in behalf of a new corporation which will be controlled by the bondholders of the present company. The receivership was instituted last fall to conserve the resources and to facilitate the reorganization. John H. Waters, president of National Radiator, will head the new corporation.

Frigidaire Units Installed in Model Home

Room cooling has not been overlooked by the designers of the Wonder House, on display in the auditorium of John Wanamaker's department store, New York, conceived and planned by *Pictorial Review* in cooperation with Wanamaker's and leading manufacturers in the building and related industries. Two Frigidaire room cooling units have been installed, one in the living room and the other in the master's den.

The Wonder House is a full-size, six-room Norman French cottage, complete and modern in every detail, designed by Bernhardt E. Muller, architect, and built by Arthur R. Rule. The house is a model of compactness and convenience, and contains the very latest in mechanical and electrical equipment, many developments being exhibited here for the first time.

The Wonder House is being reproduced in permanent form in Wychwood, Westfield, N. J. It is expected that a Petro oil burner will be installed.

New Literature

Alexander Brothers, Inc., 14 South St., Philadelphia, Pa., has issued a Power Transmission Guide for leather belt drives, catalog No. A-103. It contains 32 pages of engineering data and design instructions to meet modern industrial requirements.

The B. F. Goodrich Rubber Co., Akron, Ohio, has issued a guide to the selection of belting, hose, and other products. It contains 24 pages, is illustrated, and is standard size. The publication is entitled "Engineering Data, Industrial Rubber Goods."

Taylor Forge & Pipe Works, P. O. Box 485, Chicago, has issued a booklet, Taylor Forge, containing a general description of some of its products, and several interesting installation photographs. Seven pages.

Ingersoll-Rand, 11 Broadway, New York, has issued a folder describing its new line of Cameron Motorpumps. Includes installation illustrations and sectional view of pump and motor to show construction details.

Wagner Electric Corp., 6463 Plymouth Ave., St. Louis, has issued a six-page bulletin describing multi-speed squirrel-cage motors. Photographs of installations illustrate the bulletin. The description covers constant-torque, constant-horsepower, and variable-torque motors. Designated Bulletin 174, Part 5, pages 13-18.

York Ice Machinery Corp., York, Pa., has issued "Cold Magic," an 85-page book telling the story of the progress of refrigeration during the past half-century. It is attractively bound and beautifully illustrated with photographs.

American Radiator Co., New York, has issued "Ideal Heating for Large Buildings," a catalog describing the company's large type Redflash and water tube boilers.

Entrance and view toward dining alcove of the Wonder House, equipped with Grinnell sprinklers and Frigidaire room coolers. Cooling unit is shown to left of living room

With the Manufacturers

Westinghouse Announces Air Conditioning Division

Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., has organized the air conditioning equipment division in the supply engineering department for the development and sale of air conditioning apparatus.

J. W. Speer

J. W. Speer, formerly manager of machinery electrification, industrial department, has been appointed manager of commercial air conditioning products of the company.

W. C. Goodwin

W. C. Goodwin has been appointed division engineer in charge of air conditioning work.

Northeastern Construction Lays 8 Miles of Pipe for Government

Contract for 42,800 ft. of underground steam mains to be installed in approximately 19,000 ft. of tunnel and conduit servicing public buildings in Washington, D. C., has been awarded to the Northeastern Piping and Construction Corporation, North Tonawanda, N. Y. The steam distribution system which will cost approximately $1,250,000 was designed for the government by United Engineers and Constructors, Philadelphia. The work was started the middle of August and will require about one year for completion.

Many of the buildings in Washington, including the White House, are now heated by steam from a district heating system. When the new project is finished most of the public buildings in Washington will be serviced by underground steam mains.

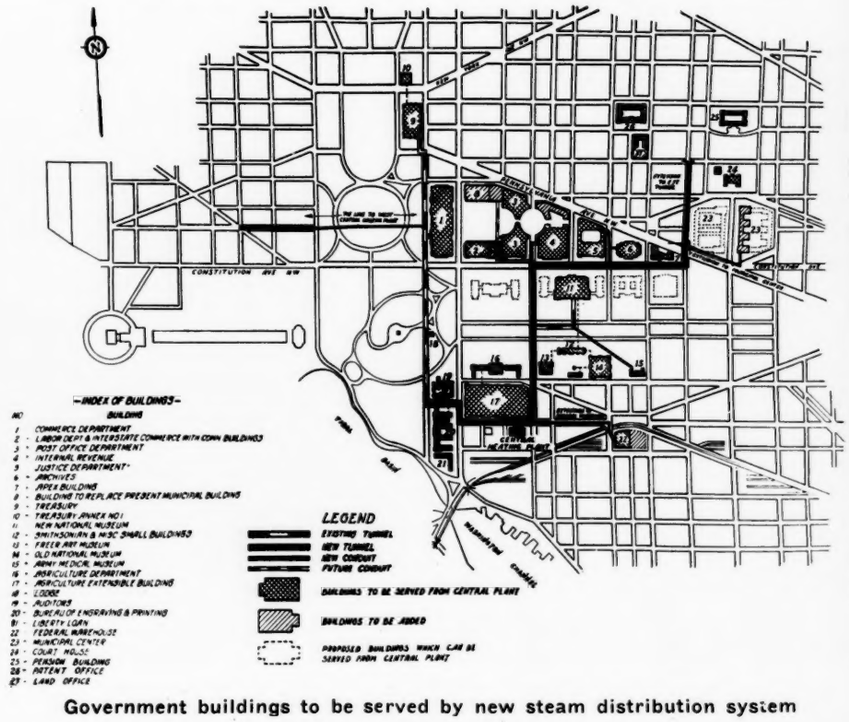
The district to be served by the new system is bounded by 6th Street, N.W., E Street, N.W., 12th Street, N.W., and C Street, S.W. The power plant, contracts for which have not yet been placed, will be located on C Street, midway between 6th Street and 13th Street—approximately in the center of the C Street boundary line.

The system will service the following buildings: Buildings planned or under construction—Commerce, Post Office, Justice, Archives, Apex building, Auditors building, and Municipal building. Existing buildings—Internal Revenue, Labor and I.C.C., Treasury, Treasury Annex 1, the National, Smithsonian, Freer, Old National, and Army Medical Museums, Agriculture, Agriculture Extension, Lodge, Printing, Liberty Loan, Court House, Patent Office, and others.

The underground steam distribution system is designed for 300 lb. working pressure and calls for 8900 ft. of new reinforced concrete tunnel 7 in. x 7 in. and 7 in. x 6 in. inside dimensions; 4800 ft. of reinforced concrete conduit approximately 4 in. x 4 in. inside dimensions and 1000 ft. of tile conduit 10 in. x 12 in. and 4800 ft. of main installed in existing tunnels. Approximately 42,800 ft. of pipe mostly from 6 in. to 18 in. will be installed in tunnels and conduit.

High pressure steam mains will range from 18 in. down to 4 in. while condensate, return and drip mains will be from 10 in. to 1¼ in.

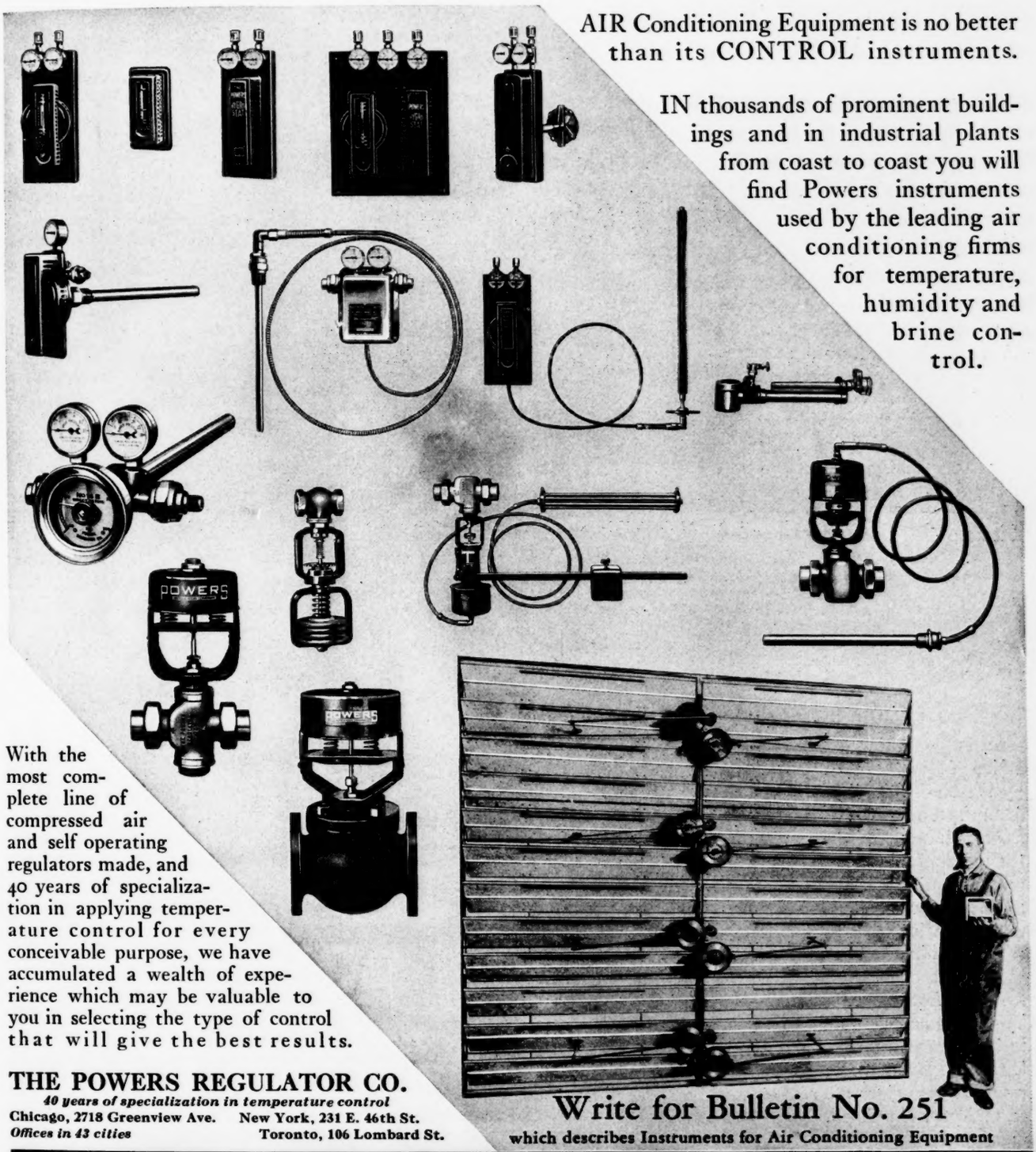
Specifications call for 246 slip type and packless expansion joints, varying from 2 in. to 18 in., numerous expansion loops, steam flow meters, and electric pumps at low points in tunnel and conduit to take care of any seepage water.



Government buildings to be served by new steam distribution system

POWERS Control

for Air Conditioning Equipment *is used by the World's greatest experts in Temperature and Humidity Regulation*



AIR Conditioning Equipment is no better than its CONTROL instruments.

IN thousands of prominent buildings and in industrial plants from coast to coast you will find Powers instruments used by the leading air conditioning firms for temperature, humidity and brine control.

With the most complete line of compressed air and self operating regulators made, and 40 years of specialization in applying temperature control for every conceivable purpose, we have accumulated a wealth of experience which may be valuable to you in selecting the type of control that will give the best results.

THE POWERS REGULATOR CO.

40 years of specialization in temperature control

Chicago, 2718 Greenview Ave. New York, 231 E. 46th St.

Offices in 43 cities Toronto, 106 Lombard St.

Write for Bulletin No. 251

which describes Instruments for Air Conditioning Equipment

POWERS

With the Manufacturers

Unit Heater Association Inaugurates Certified Ratings

At a recent meeting of the Industrial Unit Heater Association in Chicago a system of certified ratings was adopted whereby members will designate by means of the sticker reproduced here that their products are tested according to the standard test code adopted jointly by the association and the A.S.H.V.E. This action follows similar certification by the National Association of Fan Manufacturers, whose sticker also is reproduced.

CERTIFIED
RATINGS
Ratings of unit heaters are in accordance with standard test code for unit heaters adopted jointly by the Industrial Unit Heater Association and the American Society of Heating and Ventilating Engineers
MEMBER OF
I.U.H.A.
CERTIFIED
RATINGS
Air deliveries are in accordance with Standard Test Code for Centrifugal and Propeller Fans adopted jointly by the National Association of Fan Manufacturers and American Society of Heating & Ventilating Engineers.
MEMBER OF
NAFM

Certification Stickers of I.U.H.A. and N.A.F.M.

The sticker of the Industrial Unit Heater Association reproduced at the left has red and green lettering on a black background. To the right is the sticker of the National Association of Fan Manufacturers, which is on a blue background. Stickers are reproduced actual size.

Copies of either of these codes may be had gratis upon application to HEATING AND VENTILATING or to the associations' headquarters at 308 West Washington St., Chicago.

Richardson & Boynton, New York, announces that its manufacturing plants at Buffalo, N. Y., Dover, N. J., and Utica, N. Y., have resumed operations after an extended mid-year shutdown.

Petroleum Heat & Power Co., Stamford, Conn., announces the opening of a factory branch sales office at 415 Boylston St., Boston, Mass. J. Henry Brody is sales manager.

Allis-Chalmers Mfg. Co., Milwaukee, Wis., states that Walter Geist, assistant manager of the milling machinery department, including the Texrope Drive Division, will sail for Europe September 9. Mr. Geist will spend three months on the continent and the British Isles where he will study conditions and visit the company's offices in England, France, Germany, Italy, and other countries.

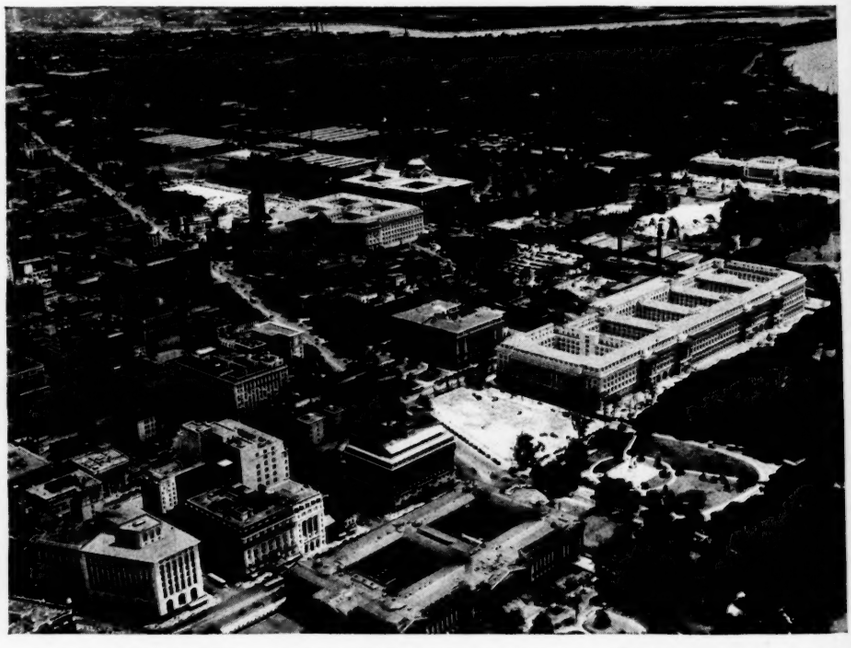
Thermoil Heaters, Benton Harbor, Mich., announces that F. L. Phillips has been appointed purchasing agent.

Pittsburgh Equitable Meter Co., Pittsburgh, Pa., announces the appointment of Fred Arnesen as manager of its Pacific Northwest territory, with headquarters in Seattle, Wash. Mr. Arnesen was formerly district manager of the company's Dallas, Tex., office.

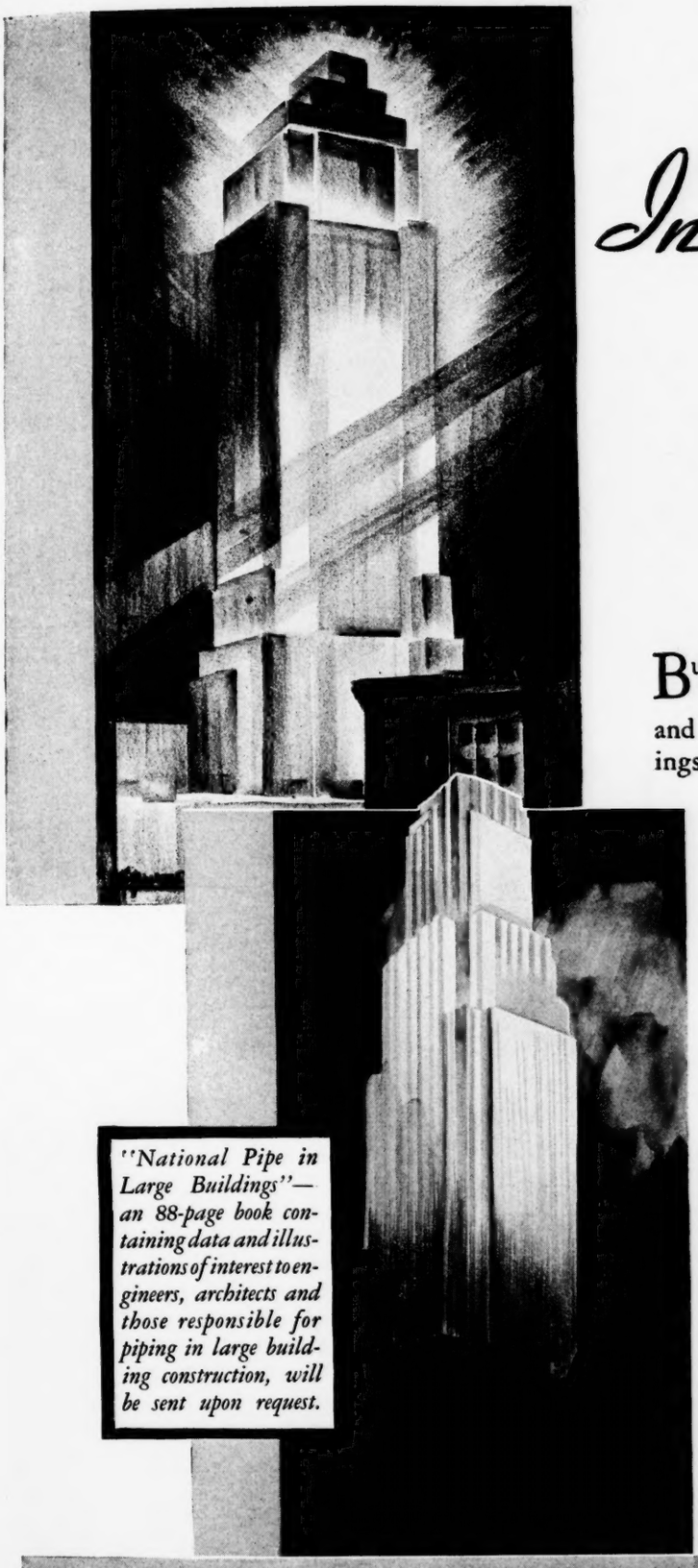
Electrol Inc., New York, has opened a new office and showroom at 423 Boylston St., Boston, Mass. Ted Walker is the Boston branch manager.

Fitzgibbons Boiler Co., Inc., New York, announces the appointment of William Heagerty, 315 Davidson Ave., Buffalo, N. Y., as distributor of its products and the products of the Kewanee Boiler Co., Inc., New York in the western New York territory and several counties in northwestern Pennsylvania.

The Silent Glow Oil Burner Corp., Hartford, Conn., has purchased the assets of The International Burners Corporation and is now manufacturing the latter's line of Caloroil, Victory and Crescent burners through its newly-formed Caloroil Burner Division. M. J. Binkley, who was vice-president of The International Burners Corporation, will be in charge of the engineering, manufacturing and sales organizations of the new division.



Air view of Washington buildings to be heated by new district heating system



In BUILDINGS *yet to* RISE!

Buildings of lofty height and advanced design make more and more strenuous demands on materials and mechanical equipment. In the wondrous buildings of the future, whatever their dimensions and whatever forms they may take, still greater requirements are certain to develop.

Throughout its history, the improvement of NATIONAL Pipe has kept pace with every progressive step in heating and piping practice. The NATIONAL policy admits of no contentment with past success at any time. It can be satisfied only with what is constantly improved. As new conceptions and new undertakings in the building field or elsewhere create new standards, NATIONAL will be ready to meet and match them with suitable products. To specify NATIONAL Pipe will always be prudent, economical, modern, in years to come as in the past, for no effort and no resource will be spared that may be needful, to deserve and to hold its place as—

America's Standard Wrought Pipe

"National Pipe in Large Buildings"—an 88-page book containing data and illustrations of interest to engineers, architects and those responsible for piping in large building construction, will be sent upon request.

NATIONAL TUBE COMPANY
Subsidiary of United States Steel Corporation
PITTSBURGH, PA.

NATIONAL PIPE

With the Manufacturers

Inland Steel Co., Chicago, announces the opening of offices at 1118-19 Fisher Building, Detroit, Mich., and the appointment of J. H. Fitch, Jr., as district sales manager.

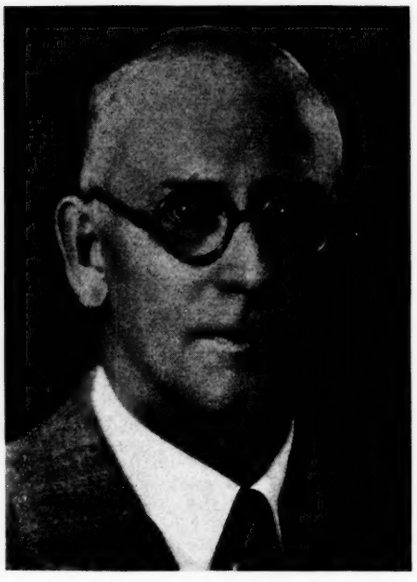
International Cooler Corp., Toronto, Canada, will commence manufacture of its products in the Westman Machinery Company plant at Oshawa. The company manufactures air conditioning equipment, electric refrigeration equipment and dairy refrigeration machinery. John T. Gowan is production manager, S. N. Dancey, vice-president, and Charles Howard, managing director of the company.

Davis Engineering Corp., New York, announces the appointment of John N. Dickson, 603 New Sprankle Building, Knoxville, Tenn., as district representative in the Knoxville territory of its line of domestic water heaters. Arthur C. Hall Co., 12 Tecumseh St., Worcester, Mass., is representing the corporation in that vicinity.

Fisher Governor Co., Marshalltown, Iowa, has established two new exclusive representatives. One is the Grinnell Co., 240 Seventh Ave., S., Minneapolis, Minn., and the other is the Steam Plant Equipment Co., 1012 N. Third St., Milwaukee, Wis.

Silent Knight Oil Burner Co., Atlantic, Iowa, has been organized as a division of the Morton Oil Burner Co., Atlantic, to market the Silent Knight oil burner, for many years sold by the Morton company. L. P. Lang is general manager, and C. J. Smith, general sales manager, of the new company.

Kleen-Kwik Chemical Co., Inc., Flint, Mich., manufacturer of a scale remover and cleaner for steam coils, pipes, heating systems, etc., has appointed Shay-West Associates, 616 S. Michigan Ave., Chicago, to handle the sale of its product and function as the sales department of its company.



Clint E. Beery

Spencer Heater Co., Williamsport, Pa., has appointed Clint E. Beery as its representative in Chicago. Mr. Beery, who was the organizer of the Mid-West Stoker Association, will be in charge of the newly-opened Spencer office in Chicago. He will also be in charge of a branch soon to be opened in Milwaukee.

Pierce, Butler & Pierce Mfg. Corp., has moved its executive offices from New York to 282 James St., Syracuse, N. Y.

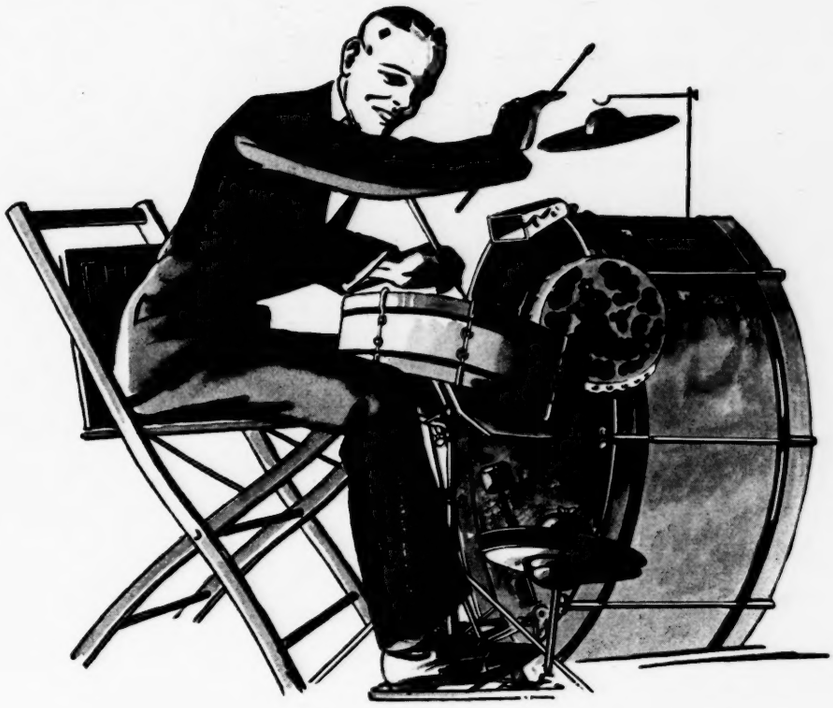
Sundstrand Eastern Sales Co., New York, has appointed John H. Hassinger, formerly sales manager for International Burners Corp., New York, as its manager. The company will handle Sundstrand sales in the east, and has started sales activities in the metropolitan New York area.

Lennox Furnace Company of Canada Ltd., Toronto, announces that the controlling interest of its company has been transferred to Canada. It formerly was controlled by the Lennox Furnace Co., of Syracuse, N. Y., and Marshalltown, Iowa. Major L. L. Anthes, president, Anthes Foundry Ltd., will be president of the newly-controlled company.

Perfex Corp., Milwaukee, Wis., has appointed O. H. Wolff, 6228 Oakland Ave., St. Louis, as its representative in that territory; the Air Equipment & Supply Company in Cleveland, and F. O. Glas in Milwaukee.

Young Radiator Co., Racine, Wis., announces the appointment of O. D. Marshall, 514 Anderson Building, Grand Rapids, Mich., as sales representative of its company's heating division in that territory.

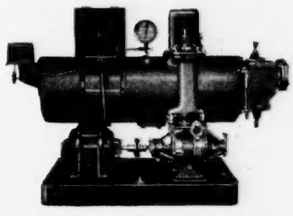
Columbus, Ohio, research and testing laboratory of the Owens-Illinois Glass Co., Toledo, Ohio, where research work is conducted on the company's line of filters



SIMULTANEOUS CAPACITY!

That quality of being able to do two things at once. A trap drummer can do it, but not all heating pumps can. A return line heating pump has to pump air and it has to pump water. Will the pump you contemplate buying pump both at once? A square foot rating on a heating pump represents nothing but the *opinion* of the manufacturer of that pump that it will handle the returns from a heating system of a certain size. The important thing is: How much air and water will it pump in a minute? After all, that is what you are buying, isn't it? So much air and water a minute pumped at a definite horse power.

Now most manufacturers of heating pumps do give a definite rating of water and air, and most pumps will pump their rated volume of air *or* of water. But what it is important for you to know is whether the pump will handle both together, at once, at the same moment. In other words, *SIMULTANEOUSLY.* Understand, not "Will it pump air and water at the same time?" Some pumps will even do that. Will it pump its *FULL RATED CAPACITY* of air *AND* its *FULL RATED VOLUME* of water and at its *RATED HORSE POWER*, and do it *SIMULTANEOUSLY?* The answer to that simple little question means a perfectly operating heating system with ample reserve capacity for zero weather, and it means a big saving in electricity. It means dollars. Make sure you have it right when you specify a pump, and you will specify a Jennings.



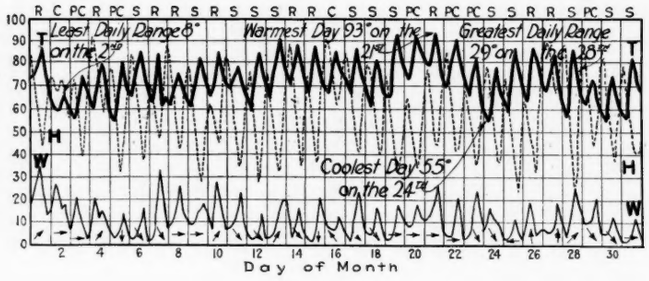
NASH ENGINEERING CO., SOUTH NORWALK, CONN., U. S. A.

Jennings Pumps

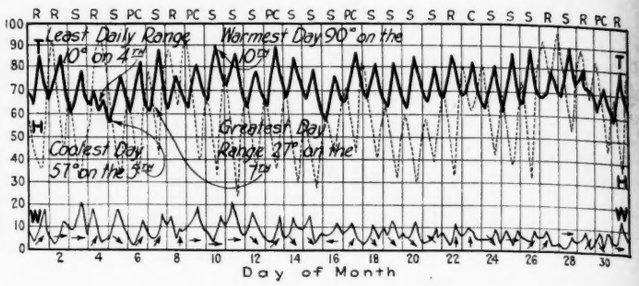
SIMULTANEOUS CAPACITY

The Weather

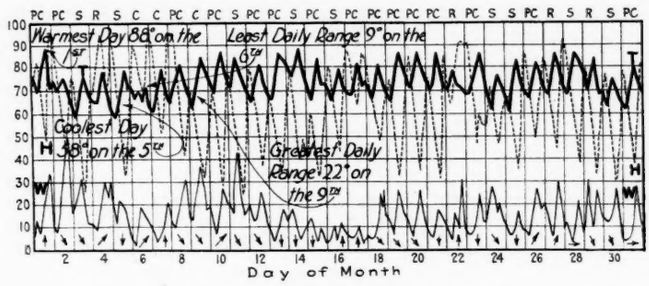
July, 1932


Least Daily Range 8° on the 2nd
Warmest Day 93° on the 21st
Greatest Daily Range 29° on the 28th
Coolest Day 55° on the 24th
Day of Month

Pittsburgh

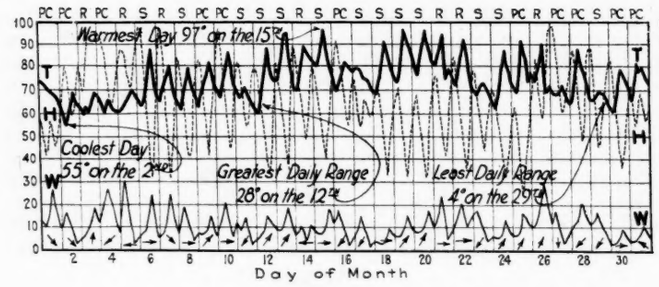


Boston

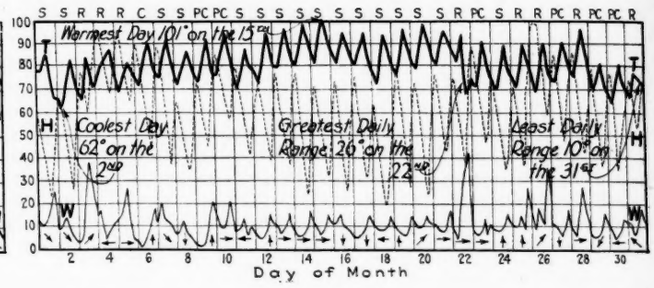

Warmest Day 88° on the 1st
Least Daily Range 9° on the 6th
Coolest Day H 38° on the 5th
Greatest Daily Range 22° on the 9th
Day of Month

New York

(Hourly observation of the Relative Humidity Plotted on this Chart)


Warmest Day 97° on the 15th
Coolest Day 55° on the 24th
Greatest Daily Range 28° on the 12th
Least Daily Range 4° on the 29th
Day of Month

Chicago


Warmest Day 101° on the 15th
Coolest Day 62° on the 2nd
Greatest Daily Range 26° on the 22nd
Least Daily Range 10° on the 31st
Day of Month

St. Louis

	New York	Boston	Pittsburgh	Chicago	St. Louis
Highest temperature, deg. F.	88	90	93	97	101
Date of highest temperature	1	10	21	15	15
Lowest temperature, deg. F.	58	57	55	55	62
Date of lowest temperature	5	5	24	2	2
Greatest daily range, deg. F.	22	27	29	28	26
Date of greatest daily range	9	7	28	12	22
Least daily range, deg. F.	9	10	8	4	10
Date of least daily range	6	4	2	29	31
Mean temperature for month, deg. F.	73.9	72.8	73.0	74.7	81.6
Normal mean temperature for month, deg. F.	73.8	71.6	74.7	72.5	78.8
Total precipitation, this month, inches	0.85	2.10	2.01	4.67	5.30
Total snowfall, this month, inches	None	None	None	None	None
Normal precipitation, this month, inches	4.24	3.49	4.05	3.33	2.98
Total wind movement, this month, miles	10,365	5,566	7,084	7,324	7,486
Average hourly wind velocity, miles	13.9	7.5	9.5	9.8	10.1
Prevailing direction of wind	N.W.	S.W.	W.	S.W.	W.
Number of clear days	10	16	9	12	18
Number of partly cloudy days	17	8	13	16	10
Number of cloudy days	4	7	9	3	3
Number of days with precipitation	7	9	14	11	9
Number of days with snowfall	None	None	None	None	None
Snow on ground, at end of month	None	None	None	None	None

Plotted from records especially compiled for HEATING AND VENTILATING by the United States Weather Bureau.
Heavy lines indicate temperatures in degrees F. Light lines indicate wind in miles per hour.
Broken lines indicate humidity in percentage from readings at 8 a. m., 12 m., and 8 p. m.
S—clear, PC—partly cloudy, C—cloudy, R—rain, Sn—snow. Arrows fly with prevailing directions of wind.

Heating and Ventilating

Engineering
Library

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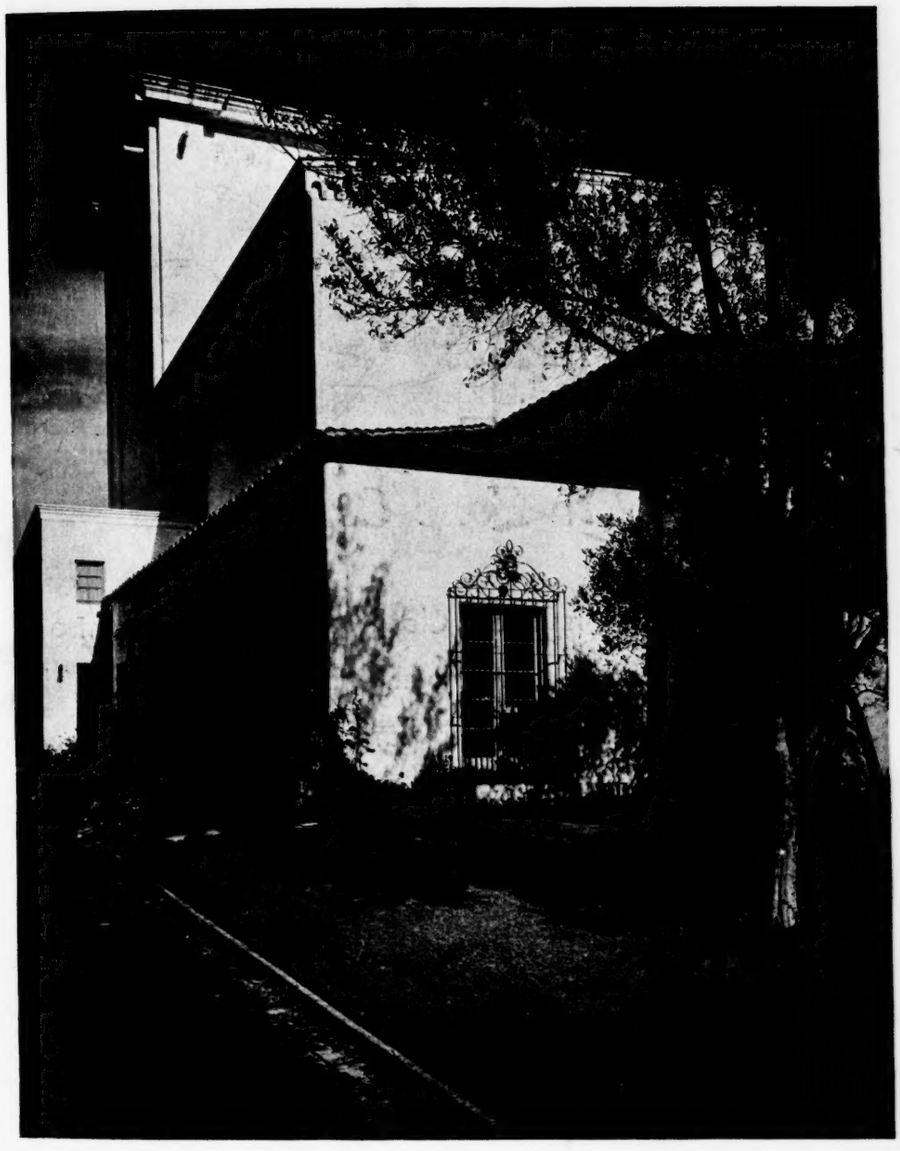

OCTOBER
1932

AIR CONDITIONING FOR SMALL BUILDINGS